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The Factors Affecting Locomotive Tractive Effort

Railway Mechanical Engineering

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THE FACTORS AFFECTING LOCOMOTIVE TRACTIVE EFFORT

BY

EVERETT GILLHAM YOUNG
B. S. University of Illinois, 1913

THESIS

Submitted in Partial Fulfillment
of the Requirements for the
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May 27, 1916

I HEREBY RECOMMEND THAT THE THESIS PREPARED UNDER MY SUPERVISION BY EVERETT GILLHAM YOUNG

ENTITLED THE FACTORS AFFECTING LOCOMOTIVE TRACTIVE EFFORT.

BE ACCEPTED AS FULFILLING THIS PART OF THE REQUIREMENTS FOR THE DEGREE OF Master of Science in Railway Mechanical Engineering

In Charge of Thesis

Edward C. Schmidt

Head of Department

Recommendation concurred in:*

Committee on Final Examination*

*Required for doctor's degree but not for master's.

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TABLE OF CONTENTS.

	page
I. The Tractive Effort-Speed Curve	1
1. The Problem	1
2. Scope of the Present Study	2
3. Previous Studies of the Problem	4
4. Interpretations of the Speed-Pull Relation	6
II. A Study of Proposed Tractive Effort Methods ...	11
1. Classification	11
2. Rational Methods	12
3. Semi-rational Methods	20
4. Empirical Methods	24
5. Comparative Plots	30
6. Summary	30
III. The Variables of the Problem	35
1. The Principle Variables	35
2. Effect of the Primary Variables on a Tractive Effort Estimate	39
3. Other Variables	43
IV. Bases of Comparison of Tractive Effort in Engines of Widely Varying Design	47
1. Pounds of Pull and Speed	47
2. Fractions of Rated Tractive Effort and Speed	48
3. Elimination of the Steam Distribution and Superheat Variable	49
4. Comparison on Evaporation Basis	51
V. Tractive Effort Variation in Superheater Locomotives	55
1. Comparison of Results of Laboratory Tests.	55
2. Relation Between Boiler Capacity and the Pull at 600 f.p.m.	65
3. The Relation for Other Speeds	67
VI. Application of the Process to Locomotives Using Saturated Steam	71
VII. Methods of Estimating Boiler Performance	79
1. The Variables of the Problem	79
2. Quality of the Fuel	86
3. An Index of Boiler Capacity	88
4. Comparison of Data on Equivalent Grate Basis	92
5. Preliminary Investigations	94
6. Method of Using Curves	99
VIII. Outline and Summary of the Process	100
1. The Process Summarized	100
2. Examples of Application	106

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TABLE OF CONTENTS (concluded).

	page
Appendix II-A	119
Appendix III-A	134
Appendix III-B	136
Appendix V-A	139
Appendix V-B	140
Appendix V-C	142
Appendix V-D	144
Appendix V-E	145
Appendix V-F	146
Appendix V-G	150
Appendix V-H	151
Appendix VI-A	154
Appendix VI-B	156
Appendix VI-C	163
Appendix VI-D	166
Appendix VI-E	167
Appendix VII-A	168
Appendix VII-B	170
Appendix VII-C	174
Appendix VII-D	175
Appendix VII-E	176
Appendix VII-F	181

THE FACTORS AFFECTING LOCOMOTIVE TRACTIVE EFFORT.

I. THE TRACTIVE EFFORT-SPEED CURVE.

1. Nature and Importance of the Problem.
2. Scope and Limitations of the Present Study.
3. Methods of Forecasting Pull at Various Speeds.
   a. Basic theories.
   b. Agreement among proposed methods.
4. Mathematical and Physical Interpretations of the Speed-Pull Relation.

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1. The Problem. The variation of tractive effort with speed in a locomotive is a problem which has been attacked many times and by many methods, and this fact in itself is sufficient evidence of its importance. Some knowledge of the relation between speed and pull is required in at least three classes of problems: those which involve the original location of a line of railway or its relocation, problems in the design of new locomotives, and problems in tonnage rating -- either the calculation of theoretical ratings or the checking of ratings already in use. These problems require tractive effort estimates of varying degrees of accuracy: in a location problem, probably an error of twenty percent would be admissible; in a grade revision problem, the estimate should be closer -- within

ten per cent. if possible. In the latter class of problems -- those of design and hauling capacity, a ten per cent. error would be too great: such an estimate would be of little more value than the guess of an experienced operating man -- and probably of less value if comparable conditions existed on which to base a mental estimate.

2. Scope of the Present Study. Strictly speaking, almost every circumstance connected with the design and operation of a locomotive has its individual influence upon tractive effort; but any attempt to produce a generalization must of necessity be based upon narrower grounds. Past studies of the question have usually eliminated weather conditions, the condition of the locomotive as regards maintenance and repair, and all variables in what might be termed the "personal equation" of the enginemen, with the exception of the rate of firing. No previous consideration of the question has taken into account other important causes of variation: the relation between grate area and heating surface,* which has a large effect upon the evaporative efficiency, or the design of the engine as to steam distribution -- that is, the efficiency of the valves, valve gears, piping and ports. In the present study, an effort will be made to evaluate the following variables:

1. Rated tractive effort (or more properly, the quantity pld^2/D)

2. Speed

3. Evaporative capacity

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* To this statement might properly be excepted the work of Mr. Shurtleff. See Section II, p. 18 , and Section VII, p. 85.

4. The design of the steam distribution apparatus.

Only the "cylinder tractive effort" will be considered, in order that comparisons may be made on a basis fair to all engines. No attempt will be made to correct results to give the pull at the tender drawbar, or to take into consideration the machine friction of the working parts of the engine. This is essentially an independent problem, and no satisfactory solution has as yet been produced. It is doubtful whether the existing data would permit of a generalization. For practical purposes the usual assumptions are satisfactory. There are two of these assumptions in common use:

$$F = 22.2\ W$$

$$F = 18.7\ W + 80\ D$$

In each case, F is the total machine friction in pounds, W the weight on the driving-wheels in tons, and D the number of driving wheels. The former assumption is merely an average value adopted by Mr. Cole and others. Recent tests have shown this value to be rather low, and the use of 25 pounds per ton is recommended by the American Railway Engineering Association in its publications. The calculation of the rolling resistance by which the cylinder tractive effort is reduced is a relatively simple matter. From "Freight Train Resistance" by Prof. E. C. Schmidt the resistance for the tender may be calculated directly,* considering it as a car of the appropriate weight. The weight on

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* Bulletin 43, Engineering Experiment Station, University of Illinois. Table 3, p. 35 is convenient for the purpose.

the engine trucks is treated in an analogous manner: for example, if a four-wheeled leading truck carries a weight of thirty tons, the unit resistance for that weight would be the same as that for a sixty ton car. Where assumptions are required herein concerning additions or deductions to be made for friction or resistance, the methods here suggested will be followed, unless actual test data are available for engines of the same design and similar weight. However, as far as possible the conclusions reached will be based upon indicator cards or figures for indicated horse power, in order to minimize errors introduced from this source.

3. Previous Studies of the Problem. The variety of principles upon which efforts to solve the tractive effort problem have been based is great, extending from a complete and rational analysis of the performance of the locomotive to a mere estimate based upon the tests of a single locomotive. The various processes may be classified for convenience as rational, semi-rational, and empirical methods, though further study shows that all have empirical elements. The classes may be further subdivided and interpreted as follows:

Rational Methods:

Constant horse power assumed, based upon heating surface.

Varying horse power assumed, determined by dividing the evaporation by the water-rate for each speed.

Semi-rational methods:

In which some or all of the features of the design are taken into account, the process being in the main empirical.

Empirical methods:

Usually attempts to produce generalizations from meager data, the product appearing in the form of a series of "speed-factors" by which the rated tractive effort is to be multiplied. Evidently if these are applied in any case other than one identical with that from which the relation was produced, no account is taken of two of the principal variables, boiler capacity and steam distribution.

As might be expected from so much variation in principle, the variations in the results obtained are very wide. Fig. 3, 4, 5, and 6 show the results given by a number of the proposed methods when applied to a particular engine. A study of the curves shows the following percentages of variation, the percentage in each case represents the amount by which the highest curve exceeds the lowest, only the rational methods being considered:

Variation for	10 m.p.h.	20 m.p.h.	30 m.p.h.
Saturated Steam Locomotives	68%	87%	120%
Superheated Steam Locomotives	33%	76%	81%

This showing in itself furnishes sufficient incentive to a further study of the subject. It is evident from the curves that particular attention should be paid to that portion sometimes called the "knee" where the ordinates begin to decrease rapidly, for in the case of freight engines an accurate knowledge of the power available is of the greatest importance within the range of speed at which ruling grades are generally surmounted.

4. Interpretations of the Speed-pull Relation. The relation between pull and speed is not one which readily lends itself to a purely mathematical expression, the curve taken as a whole being unlike any of the usual functions. It is possible that an adaptation of the form of the cubical parabola might be made, the position of the origin being fixed by the relation between boiler capacity and related tractive effort, and variations of other kinds shown by predetermined methods of modifying the coefficients. Another curve and probably a more simple one is the "witch", though this has the disadvantage of becoming horizontal, as shown.

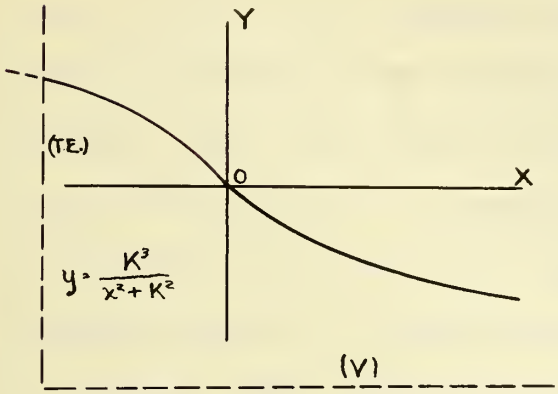

Fig. 1. The Cubical Parabola.

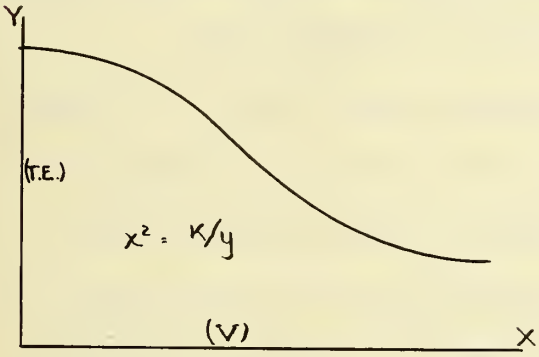

Fig. 2. The Witch.

This difficulty has been met in another way -- by determining the forms of the low-speed and high-speed parts of the curve separately. By "low-speed" section is meant that part of the curve which is nearly horizontal, below the speed at which the pull rapidly decreases. The high-speed portion then approximates a hyperbola, and the low-speed section a straight line: this mathematical form was used in the first rational solution of the problem which was published in this country.* There are serious objections to this form of

* See the "Goss method", Section II, p. 12.

treatment, based upon its inaccuracy. Any mathematical statement of a relation between speed and tractive effort would involve in addition the evaporation, and the cylinder efficiency, or water rate. The evaporation in turn is a function of the coal fired both as to quantity and quality, and the boiler design, chiefly the tube length and the proportion of firebox or direct heating surface to tube surface. The water rate is a function of cylinder size, cut-off, steam distribution (i.e., valves, gear, pipes and ports) and speed. Here then are four main and eight subordinate variables to be taken into account in an equation. By making certain well founded assumptions a number of these variables may be removed, and a form of equation suitable for the high speed range at least may be developed. This has been done and the results will be shown later.*

The physical interpretation of the curve is much simpler. At the start, pressure is "piled up" in the cylinders; if the load is a tax on the capacity of the engine, the pressure in the cylinders will be equal to that in the boiler. As soon as the piston is set in motion it becomes impossible to attain full boiler pressure, except probably instantaneously in the clearance space when the piston is just past the dead center; even under these conditions wire-drawing will somewhat reduce the pressure of the steam in the cylinder itself. The mean effective pressure of the first stroke will be further reduced by the fact that "full cut-off" -- that is, admission continuing to the end of the stroke -- is impossible, the steam being

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* See the "Kiesel Formula", Section II, p. 22.

forced to expand through the last ten or twelve per cent of the piston-travel. As long as the steam supplied by the boiler is ample, the mean effective pressure would tend to remain constant, but the effect of wire-drawing, which from the laws of fluid friction increases with the square of the velocity of flow, causes an increasing rate of falling off. A speed is soon reached when the demands of the cylinders tax the boiler capacity, though this is postponed by the increase of evaporation which takes place with the rapid increase in the rate of combustion and the draft; at this point the boiler pressure begins to fall slightly, and the reverse lever is "hooked back" and the cut-off shortened. The quantity of steam used per stroke and hence its velocity through the ports and passages is decreased and consequently the wire-drawing effect is also reduced, and this partially balances the decrease in mean effective pressure due to the shorter cut-off and longer expansion, so that the decrease in mean pressure is not abrupt. Further increase in the speed requires a further decrease of the cut-off, and the amount of tractive effort depends largely on the ability of the boiler to supply steam. The effect of the size of the cylinders at these higher speeds shows only in the efficiency. A given speed and load require a certain cut-off in the cylinders. If this is other than the most economical cut-off, the steam supply is in effect reduced, since more steam is required to perform a given amount of work.

At high speeds the variation of the pull is best studied through the medium of the variation of horsepower. In

the case of the superheater locomotive, the steam consumption becomes almost constant at speeds above 1000 feet per minute piston speed. Since the evaporation varies but little, the indicated horsepower is practically constant; hence the speed-pull curve is a hyperbola, since P x V x a constant is equal to a constant, the horsepower. In the case of the saturated engine, the water rate or steam per indicated horsepower hour reaches a minimum at some point generally between 700 and 1000 feet per minute piston speed, and thereafter rapidly increases, so that the horsepower must correspondingly decrease, and the curve will lie below the hyperbola at all points, the ordinates becoming zero at a definite point. The point of intersection of the curve with the curve representing the resistance of the engine and its tender in either case is of greater interest than the zero-point. It is certain that the superheater locomotive could propel itself at a speed far higher than the governing mechanical condition (counterbalance effect) would permit, and it is probable that the saturated steam locomotive could accomplish the same thing though the highest attainable speed would doubtless be much lower. Calculations made several years ago* showed that at 100 miles per hour a certain Atlantic type locomotive could just propel itself; if this were true and mechanical limitations permitted, a superheater locomotive of equivalent design should

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* "The Resistance of Locomotives at High Speeds", Railroad Gazette, Vol. 37 (1904), p. 166.

be able to reach 150 miles per hour or more.*

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* In this connection it is of interest to note that engine 999 of the New York Central is credited with having run at a rate of 112.5 miles per hour on one occasion, hauling a three car train. The writer has seen a model locomotive of 4.5-inch gage having 6-inch driving wheels run 22 miles per hour, or about twelve hundred revolutions per minute! The owner of the model claimed that it could run 48 miles per hour; this was seriously doubted, however, though at the former speed the safety valve was in operation a large part of the time.

II. A STUDY OF PROPOSED TRACTIVE EFFORT METHODS.

1. Classification.
2. Rational Methods.
   a. The Goss method.
   b. Houston's modification.
   c. Williamson's method.
   d. Houston's modification.
   e. The A. R. E. A. method.
3. Semi-rational Methods.
   a. The Baldwin curves.
   b. Kiesel's formula.
4. Empirical Methods.
   a. Henderson's (Master Mechanics') method.
   b. Henderson's hyperbola.
   c. Henderson's method (Locomotive Operation).
   d. Old Baldwin method.
   e. Old Schenectady method.
   f. University of Illinois method.
   g. Adams and Isascs method.
   h. Cole method (Am. Loco. Co.)
5. Comparative Plots.
   a. Rational methods, saturated steam.
   b. Semi-rational methods, saturated steam.
   c. Empirical methods, saturated steam.
   d. All methods, superheated steam.
6. Comparisons and Conclusions.

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1. Classification. In the last section an effort was made to point out some of the methods of attacking the tractive effort problem, and it is proposed here to consider these methods in further detail. The classification advanced will be adhered to for convenience in presenting the processes which have been built up by the various students of the question. Those methods which are based upon rational principles deserve

the first consideration in the nature of the case, and will, therefore, be first analyzed.

2. Rational Methods. The first of the rational methods published in this country was that of Dr. W. F. M. Goss.* The data used were taken from the tests of an eight-wheel locomotive having 17 x 24-inch cylinders in the testing laboratory of Purdue University. Basing the assumptions upon the average performance of this locomotive, and assuming that the evaporation per square foot of heating surface per hour is 12 pounds of water, the steam per indicated horsepower-hour is 28 pounds; it is deduced that one square foot of heating surface worked at this rate will produce 12/28 or 0.43 horsepower. From the familiar formula $HP = PV/375$, where HP is the horse-power, P the tractive effort, and V the speed in miles per hour, we have also $P = 375\ HP/V$. Where H is the heating surface in square feet, we have

$$P = (0.43\ H \times 375)/\ V = 161\ H/V$$

This is the equation of a hyperbola, having a value of infinity when the speed is zero. To meet the physical limitations upon tractive force, the higher part of the curve is discarded, and the relation completed to zero-speed by drawing a straight line at a height representing the adhesion limit of the engine. This line is extended until it intersects the hyperbola.

Dr. Goss says of this formula ".....This value will for the present be accepted as representing the normal performance of a modern engine under its usual range of action. It is not

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* See Proceedings of the New England Railway Club, Dec. 3, 1901, and also "Locomotive Performance" by Dr. Goss, Chapter XXIV.

the minimum ..... nor is it the maximum* ..... It does not apply to any particular engine, or any specific condition of running. It is merely an approximate measure of the maximum power that will be developed by a fairly representative modern engine under ordinary conditions of service."

The assumption of constant horsepower, and that close to the maximum which could be expected from the engine, produces the result expected. In the speed range through which the horsepower of a saturated steam locomotive actually is the maximum, i.e., at piston speeds from 700 to 1000 feet per minute, the curve agrees quite well with other methods of estimating; but at higher and lower speeds it is much too high. Even at the middle speed-range, the curve is higher than would be expected under any but exceptional conditions. This is in no sense derogatory to the method, since the author intended it to represent very good service conditions.

In a study of the same subject, Mr. H. A. Houston** suggested modifications applicable to this formula to make it more adaptable to the average service conditions. These modifications reduce the coefficient of H/V to 143, in place of 161, which is done by raising the water-rate to 32.5 pounds and lowering the evaporation to 11.5, for 13000 B.t.u. coal.

Mr. A. S. Williamson published in 1912*** a study of the tractive effort problem, which was an attempt to produce a

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* That is, not the absolute maximum for the Purdue engine.
** Thesis for degree of M.S., University of Illinois, 1913.
*** Railway Age Gazette, Mar. 22, 1912, p. 685.

generalization based upon the tests at St. Louis and those of the Purdue laboratory. For simple engines, there were three sets of tests available: those of two consolidation type engines tested at St. Louis, and the Purdue tests. From these three tests, a composite water-rate curve was produced, the empirical equation $S = 1500/R + .09\ R$ representing the result, in which S is the steam per indicated horsepower-hour and R the number of revolutions per minute. After consideration of the curves for evaporative capacity developed by Mr. Shurtleff in his investigation for the American Railway Engineering Association*, Mr. Williamson discards them for the simpler assumption that the evaporation will equal 0.68 pounds of water for each thousand B.t.u. in the coal used, for each square foot of heating surface, or $E = .68\ BH$. The heating surface is based upon the fire side of the tubes. Then using the formula employed by Dr. Goss we have (since $T = \frac{375\ H}{V}$, and $H = E/s$, the evaporation divided by the water rate) $T = 375\ E/SV$, where V is the speed in miles per hour, T the cylinder tractive effort in pounds, and the values for E and S are as stated.

It will be noted that in another place in his article, Mr. Williamson states that the evaporation will be assumed to be 0.68 pounds per thousand B.t.u. per pound of coal. From this and the preceding, it will be seen that he anticipates the firing of one pound of coal per hour per square foot of heating surface, though the statement is nowhere explicitly made. This makes it

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* See discussion under A.R.E.A. method, Section II, p.18.

evident that his method of calculation of the evaporation is based on the assumption that the boiler efficiency remains practically constant under average operating conditions. Since the constant 0.68 gives actual evaporation, it is easy to compute this assumed efficiency. The heat units contained in a pound of steam at 200 pounds pressure above feed water at 60 degrees, the quality being 99%, are approximately 1160; and therefore if the boiler were 100% efficient, 1000 B.t.u. should evaporate 0.86 pounds of water; whereas if in reality the evaporation is but 0.68 pounds, the efficiency is 79%. This seems to be a rather generous assumption.

Mr. Williamson lays special stress in his article on the fact that he is only selecting a set of representative conditions, and that users of the method should make their own assumptions as to evaporative capacity and water rate. An increase of about ten per cent in the evaporation estimate raises the ordinates of the curve in the same ratio, and renders the values obtained comparable with other tractive effort estimates. An interesting feature of this method of prediction is the mathematical relation* between speed and water-rate which is employed. During the medium speed range it reaches a minimum value of about 23.25 pounds (at 125 R.P.M.), while the value approaches infinity at very low speeds. The minimum given would be considered as extremely good for testing-plant performance, and seems too low an estimate for road service. If the coefficient .09 were replaced by .11, the minimum value would become 25.75, which is

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* The relation is $S = \frac{1500}{R} + .09\ R$, where S is the water rate, and R the speed in R.P.M.

probably as good a performance as may be expected on the road. The infinity value for zero-speed may be questioned, but there is in its favor the fact that the use of the very high values at the lowest speeds puts this curve into its proper geometrical form, an advantage that no other rational method possesses. In so far as the size of the cylinders and the rated tractive effort are not considered at the lowest speeds, and the curve obtains its form as stated, it ceases to be strictly rational.

With the effort to eliminate the very objections suggested, Mr. H. A. Houston has made certain modifications in the Williamson formula.* He assumes that a fireman can shovel coal at the rate of about 5000 pounds per hour for a reasonable period, such as that required to surmount a not excessively long ruling grade. This 5000 pounds will obtain the maximum assumed evaporation rates for an engine with 2860 feet of heating surface or less,** these rates being as follows:

B.t.u. value of coal Fired (Dry).	Maximum Equivalent Evaporation per square foot of heating surface per hour.
15000	13.00 pounds
14000	12.20 "
13000	11.50 "
12000	10.90 "
11000	10.35 "
10000	9.90 "

In case the heating surface is larger than the 2860 square feet, the fireman can fire less than 1-3/4 pounds of coal per square foot of heating surface per hour upon which the estimates are based, and the evaporation rate must be decreased in direct

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* Thesis, 1913, also in Railway Age Gazette, Vol. 55 (1913) p.455
** Slightly different from author's statement, for the sake of generalizing, but results will be identical.

proportion. Mr. Houston's assumption for the water-rate of saturated steam engines is given by the equation

$$S = 34.0 - 01 + 0.001R^2/4$$

the notation being as before. This is actual steam per indicated horsepower, and as the equivalent evaporation rates are given, the evaporation factor 1.207, representative for saturated steam locomotives is used. We have, since HP = H x E/1.207S,

$$T = 375HP/V = \frac{375\ H\ e}{1.207\ V\ S} = \frac{310.7\ H\ e}{V\ S}$$

Where e is the equivalent evaporation rate per square foot of heating surface per hour, H the heating surface, V the speed in miles per hour and S the water rate. The equation as given is that of a hyperbola. The relation is completed by the use of a horizontal line of the value of $.85pld^2/D$ from zero speed to the point at which the hyperbola is intersected. The data used by Mr. Houston were drawn from the Purdue, St. Louis and Altoona tests.

Both of the foregoing processes may be adapted for use with superheater locomotives. A relation between speed and water rate is required, which may be obtained from the bulletins of the Pennsylvania Test Department or elsewhere. Where a factor of evaporation is required, 1.300 may be used as representative. The selection of the proper heating surface to be used in calculating the evaporation by one of these methods is open to question. On the basis of evidence produced elsewhere the writer favors the use of the superheating surface as if it were a part of the evaporating surface in making an evaporation estimate.

In 1910 the Committee on Economics of Location of the

American Railway Engineering Association (hereafter referred to as the "A.R.E.A.") presented a method of calculating the tractive force at any speed which had been produced by Mr. A. K. Shurtleff from test-plant records and road tests. This is the most completely rational method in use for calculating the speed-pull relation, and takes into account the largest number of variables of the problem.* The method is as follows:

A rate of firing of 4000 pounds of coal per hour is assumed in the absence of definite information as to the actual amount, and the rate of firing per square foot of heating surface per hour is calculated. A table is given from which the actual evaporation rates per pound of coal may be taken for any grade of coal commonly used and for any rate of firing from 0.7 to 2.5 pounds of coal per square foot of heating surface per hour, and from this the evaporation rate and the total evaporation per hour and per minute may be figured. For various working pressures and cylinder diameters the steam used per foot of stroke is given in a second table, from which the steam used per revolution may be calculated; and by combining this with the evaporation, the number of strokes per minute which can be attained at full cut-off before the boiler pressure will begin to fall is determined. This critical speed is called "M". Tables showing the variation of water-rate with speed, the unit being M, are given. Knowing the water rate at M and the steam available

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* See the "Manual of Recommended Practice" of the Association; 1911 ed., p. 427; 1915 ed., p. 526. Also the Proceedings, Vol. 12, pt. 1, p. 631, 709; and Vol. 15, p. 138.

per hour, the horsepower and hence the tractive effort at the speed M may be calculated, the formula $T = 375\ HP/V$ being used. In the same manner the tractive force for any other speed may be obtained, starting with the water-rate for that speed in terms of M and dividing this into the total steam to obtain the horsepower. A table giving a less cumbersome method is provided, however, from which the tractive effort for all other speeds may be calculated after the tractive effort at M is known. If the calculated values are higher than the adhesion limit, the curve is completed by the use of a horizontal line having the value of one-fourth of the weight on the drivers, extending from the zero-speed point to the point where the curve is intersected.

As has been said, this is the most rational and logical tractive-effort method in use, and takes into account more of the variables than any other. It was originally proposed by Mr. Shurtleff to make the evaporation estimate by means of a chart which took into account the relation between grate area and heating surface, but this was abandoned in favor of the simpler method. The rate of firing, 4000 pounds of coal per hour, may be varied and the results respond accordingly. A more detailed study of this method of calculating the speed-pull relation and its working will be given in Section III. The A.R.E.A. method has been also adapted for use with superheater locomotives by the use of suitable supplementary tables, the data for this part of the work having been drawn from the reports of the tests made by the Pennsylvania Railroad at Altoona.

3. Semi-rational Methods. A semi-rational method is one in which the effects of certain features of the design have been to some extent taken into consideration, but in which the mathematical process employed in the construction of the estimated speed-pull curve bears no direct relation to the physical process taking place within the locomotive itself. Such methods have resulted from attempts to reconcile discordant data by taking into account other variables. The tests of the value of such a method are in the consistence of the results with the plan upon which the generalization is made, and in the final results given by the application of the process.

There are two of these semi-rational methods now in use: the Baldwin method and the Kiesel formula. In the former case there is no statement of the test-basis for the curves, and no attempt to present the results mathematically. The method recommended by the Baldwin Works consists of a set of curves* showing the relation between percentage of rated tractive effort which can be developed at the rim of the driving wheels, and the speed in miles per hour. There are several curves, each one appropriate for use with engines having a particular relation between their rated tractive force and heating surface. The following statements accompany the presentation of the method in "Locomotive Data":

> "For practical purposes it (i.e., the steaming capacity) may be taken as directly proportional to the total heating surface.... The curves assume that at high speeds one horsepower can be developed at

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* "Locomotive Data", published by the Baldwin Locomotive Works, 1914 edition.

> the rim of the drivers for each two and one half square feet of heating surface, and they allow for a lower efficiency at slow speeds... ..It is essential that the ratio of grate surface be properly proportioned for the quality of fuel. It is also assumed that sufficient fuel may be fired to enable the steam production to be pushed to the limit set by the heating surface".

As the curves are presented they apply to saturated steam locomotives only, but the substitution of the "equivalent heating surface" is recommended as a method of adapting them to locomotives using superheated steam. This "equivalent" is not the value commonly used, that is, the water evaporating surface plus one and one-half times the superheating surface, but is obtained by multiplying the water evaporating surface by 1.6 to take into account the more efficient use of the steam.

If it is considered that with saturated steam the water rate will be thirty pounds per indicated horsepower-hour, the assumption of two and one half square feet of heating surface per horsepower requires an evaporation of twelve pounds of water per square foot of heating surface per hour. In the case of the superheater locomotive, the arbitrary increase of the evaporating surface is equivalent to decreasing the amount required per horsepower to about 1.6 square feet, the corresponding water rate being 19 pounds.

The curves are presented with no statement of conditions to which they are particularly applicable or reservations as to their use, excepting those previously quoted. Under these conditions the only basis for testing the formula is its agreement with the performance of a known engine. This kind of comparison is shown in Fig. 4.

Mr. W. F. Kiesel of the Pennsylvania Railroad has developed a mathematical formula* for the variation of tractive effort with speed. This is in the hyperbolic form

$$t = \frac{K}{1 + CV}$$

where t is the tractive effort at the rim of the drivers, K and C are constants dependent on the design of the engine, and V is the speed in miles per hour. In the formula Mr. Kiesel takes account of the following factors: rated cylinder tractive force, total heating surface, evaporation rate, and the weight per cubic foot of steam under the condition of initial pressure. The formula is classed as semi-rational as no method appears by which it may be built up by consideration of the physical process of the production of tractive effort. It is rather an empirical generalization of results that have actually been obtained.

Professor Wood is directly responsible for all of the statements here quoted, as the author has not published his formula, and it is not known whether the assumptions, limitations, etc., are those which he himself would have used. The principle assumed value is that of the evaporation, which is given as follows:

For long runs, 10 pounds per square foot of heating surface per hour.

For two-hour runs, 12 pounds per square foot.

For runs of an hour or less, 15 pounds per square foot.

The water-side of the tube surface is used. The formula itself

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* "Locomotive Operation and Train Control" (A. J. Wood) p. 27.

makes the assumption that the total evaporation varies directly with the heating surface.

It will be noted from its mathematical form that as the speed increases the equation becomes more and more nearly that of an equilateral hyperbola. Under these conditions its form would be $tV = K'$, and the horsepower would also become constant. Having the hyperbolic form, the equation does not give reliable results at speeds below that at which the use of steam at full cut-off begins to exhaust the boiler. To complete the curve for low speeds, the line representing the rated tractive effort ($.85pld^2/D$) must be drawn, extending to the point at which the hyperbola is intersected, and the knee rounded off. Since the second term of the denominator varies directly with the weight of the steam per cubic foot the gain due to superheating is partly accounted for. This is also further taken into consideration by using in place of the water evaporating surface alone the evaporating surface plus the superheating surface.* Prof. Wood states that "the Kiesel results apply to conditions where the locomotive is kept in repair and worked under favorable conditions on the road".

The Kiesel curve is shown in Fig. 4 compared with a known performance under good road conditions.

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* Prof. Wood says that to calculate the relation for a superheater locomotive, the heating surface should be taken as the evaporating surface plus 1.5 times the superheating surface. However, in actual calculations he makes use of the value as given above. See p. 30 of "Locomotive Operation and Train Control".

4. Empirical Methods. The first published studies of the problem of tractive effort variation were of the empirical class: that is, a means of calculating the coordinates of the speed-pull curve was provided in the form of a set of "speed-factors", or a "mean-effective-pressure curve", the theory and the use of the two methods being identical.

Since the tractive effort at any speed is a function of the mean effective pressure available in the cylinders, a curve or set of coordinates giving the relation between mean pressure and speed may, by the multiplication of the ordinates by a constant value, serve to define the relation between tractive effort and speed. In practice, a curve has commonly been presented giving the relation between speed (in miles per hour, revolutions per minute, or piston speed) and fractions or percentages of the boiler pressure or, what was exactly equivalent, fractions of the theoretical or rated pull which could be developed at that speed.

The majority of these methods are proposed without limitations as to their use, or any statement of specific conditions to which they are applicable. This being the case, it is obvious that the general use of a single set of speed-factors assumes the same relation between boiler-capacity and cylinder-size for all engines, and further assumes that variation in the design of the parts which effect the distribution of the steam will not affect the performance. These criticisms, however, are more pertinent now than at the time when many of the empirical methods were published. Then there was no such variation in boiler capacity as now exists, and all locomotives had steam-

distribution gear of almost identical design.

There have been two general methods used in building up these speed-factor methods: first, by direct experiment, usually with a single locomotive, in rare cases with more, and second, by plotting the results obtained by previous investigators, and deducing from them an average curve. There are a large number of these tables and curves in existence, but only a few of the representative and better known ones will be considered.

Probably the first such method published in this country was that presented to the Master Mechanics' Association by Mr. G. R. Henderson in 1898.* The data upon which this was based was derived from the report of the Committee on Valves which was presented to the 1896 convention, this data having been produced mainly at Purdue University. The curve assumes that a maximum of eighty percent of the theoretical tractive force is available in the cylinders up to speeds of sixty revolutions per minute. From this point the tractive effort falls off abruptly. Multiplying the ordinate by the abscissa at several points on the curve, it is seen that the horsepower gradually increases even at the highest speeds (350 R.P.M.) which is not in accord with the results of later tests.**

In 1904 Mr. Henderson presented a second curve**, this being an average of seven other investigations, including his own former study and a uniform-horse-power curve similar to that previously referred to as having been the work of Dr. Goss at

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* Proceedings, Vol. 31, p. 52.
** See "Locomotive Operation", (G. R. Henderson) p. 377-378.

Purdue University. The other data had been produced by various experiments conducted by the Chicago & Northwestern, Burlington, Union Pacific, Southern Pacific and the American Locomotive Company. This curve is more in accord with theory, the horsepower reaching a maximum at about 130 R.P.M., and falling off slightly after 220 R.P.M. It is of interest to note that at the speed of 210 revolutions, the difference between the highest and lowest curve from which Mr. Henderson derived his average is 75% -- the highest giving 35% of the rated tractive force and the latter only 20%. The author says of this curve: "....(It) represents the average well designed modern locomotive.....If the boiler were smaller in proportion to the size of the cylinders, the drop would occur earlier; if larger, later." (The abrupt decrease occurs at 60 R.P.M. as in the author's 1898 curve).

As a convenient method of figuring tractive force, Mr. Henderson also recommends the use of the equilateral hyperbola TR = 6000, T being the per cent of the rated tractive force and R the number of revolutions per minute.* This is a "constant horsepower" curve.

For many years the "Records of Recent Construction" published by the Baldwin Locomotive Works gave a curve for calculating tractive effort. This was expressed in terms of the relation between mean effective pressure as a fraction of boiler pressure, and revolutions per minute. No basis for the curve has been presented, so far as can be learned, the mere statement

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* See "Locomotive Operation", (G. R. Henderson), p. 377-378.

being made that it is based on authoritative tests. It is seen from the curve that up to 20 revolutions, the tractive effort is practically constant, and then begins to fall off, the decrease being regular until a speed of 170 R.P.M. is reached, after which the decrease is small. The horsepower relations show that this curve is very much in error. The horsepower increases up to about 140 R.P.M., then decreases till a speed of 200 revolutions is reached, after which it again increases.

The Schenectady Locomotive Works used a formula concerning which the same remarks as to source, etc., are applicable. The form of the curve is much different, however. This curve assumes a constant cylinder tractive effort of 85% of the theoretical pull up to a piston speed of 250 feet per minute, after which the curve falls off abruptly. The horsepower relation for the curve is more consistent, the power increasing up to a piston speed of 600 feet per minute, after which it falls off very slightly.*

In 1901 the Mechanical Engineering Department of the University of Illinois conducted a series of very careful road tests, investigating the performance of four engines of the Illinois Central Railroad. From a plot showing the relation between the percentage of tractive effort which was developed at various piston speeds for each of these engines, an average curve was drawn. Three of the four curves are very close to the composite curve. On account of the fact that these engines

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* Both the Baldwin and A. L. Co. curves are published in convenient form in the Proc. of the Am. Ry. Eng. Ass'n., Vol. 11, part 2, p. 1223.

were worked to their practical limit, the curves represent exceptional road performance with coal similar to that used in Illinois. The horsepower in this curve attains its maximum at about 600 feet per minute piston speed, remains almost constant up to a speed of 1000 feet after which it slowly falls off. This is perfectly consistent with the results of the most careful later tests.

In a paper on "Tonnage Rating" before the American Railway Engineering Association* Messrs. Adams and Isaacs (of the Southern Pacific Co.) present several curves, and adopt one which is a straight line average of the five they show. This assumes that 95% of the theoretical tractive effort can be developed in the cylinders at starting, the ratio decreasing uniformly up to a piston speed of 1200 feet per minute at which the pull is 18 per cent of the rating. The straight line is adopted partly because of its convenient mathematical expression, and partly because of the wide variation of the other curves, rendering it as acceptable an average as any. The lowest curves exhibited are the Baldwin and Master Mechanics', the highest, the experimental results from a Southern Pacific engine, probably oil-fired, therefore not a fair comparison. At 750 feet piston speed, the lowest figure is 27% and the highest 55% of the value pld^2/D. The maximum horsepower is attained at 700 feet per minute, after which the decrease is rapid -- probably too rapid, since at 1200 feet per minute the power is only 45% of that for 700 feet. Through the medium speed range, the line represents a conservative

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* See reference on preceding page.

estimate for an oil-fired engine of the type shown in the plot.

Both the Baldwin and American Locomotive Companies have recently adopted new plans for estimating tractive effort. That of the Baldwin Company is a semi-rational method which has been previously discussed. That of the American Locomotive Company is a speed factor plan, as before.* This set of factors is based "on a large number of tests run under widely varying conditions" according to Prof. Wood. Two curves are given at piston speeds above 600 feet per minute, one for superheated engines and one for those using saturated steam. The assumption that the superheater has no effect on the tractive effort at speeds below 600 feet per minute** will not bear close inspection. The curve being an average one, it may be assumed that an average relation between boiler capacity and tractive effort is represented, and this proves to be the case. The agreement of this formula with the actual performance of an engine having an evaporation in pounds per hour approximately equal to its theoretical tractive effort in pounds is rather close. This however is an exceptional engine; for the average passenger engine has much higher relative boiler capacity, hence the curve is low. The average freight engine has a lower relative boiler capacity, with the result that the curve is much too high.

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* Published in Bulletin 1017 of the Amer. Loco. Co., also in "Locomotive Operation" by A. J. Wood, p. 26.

** Six hundred feet per minute piston speed is equivalent to twenty miles per hour on a freight engine with small drivers and a long stroke, and to thirty miles per hour on a passenger engine with the opposite conditions.

All of the eight curves mentioned will be found plotted in Fig. 5, together with an actual performance curve for reference. The calculations will be found in Appendix II-A.

5. Comparative Plots. Figures 3, 4, 5, and 6 have been previously referred to in the text. These figures show the results of applying the various processes of estimation to an actual case, and the known performance of this engine furnishes a check on the estimates. In each case the best possible assumptions are made to fit the conditions, as directed by the author, in order that the curves may be fairly comparable.

6. Summary. The showing of the four figures referred to makes it possible to summarize the results as follows:

The Goss method represents a high estimate throughout its range. Being characterized by a sharp knee and a very abrupt decrease of the ordinates thereafter, its use can not be recommended. Mr. Houston's modification gives a more conservative estimate, but the still sharper knee is objectionable.

The Williamson formula gives a correctly shaped curve, but with very low ordinates at low speeds. A reconsideration of the estimates of evaporation and water-rate would render the process very useful. Mr. Houston's modifications are in line with the improvements suggested, and the results are more satisfactory. However the hyperbolic form assumed by the curve results in a sharp knee caused by the use of the horizontal limiting line.

The A. R. E. A. method is most satisfactory in its mathematics, and accounts for a large number of the variables.

The results represent a conservative estimate, or what might be termed "all-day" performance. A slightly modified knee is present, and the decrease in the ordinates thereafter is too rapid. With proper judgment in its use, this method should prove an excellent tool except at speeds immediately around the knee.

The Kiesel method is flexible, leaving room for at least one important assumption to fit the case in question. This fact may be an advantage to the mechanical engineer, but to the man in the operating or the engineering department it may be the reverse. This is a hyperbolic curve, and hence would normally have a limiting line and a sharp knee; however, the directions for the use of the method specify that the intersection is to be "rounded off". This improves the shape of the curve, but does not add to the definiteness of the value of the ordinates around the knee. The formula gives very good results at high speed.

The new Baldwin method gives a curve of the correct form and makes allowance for the variation of the heating surface, though the latter is accomplished by a method open to question. A satisfactory and simple means is provided for making a rough estimate of the pull at any speed. The older Baldwin curve must be considered as entirely unsatisfactory.

The method used by the American Locomotive Company (the Cole factors) gives satisfactory results when applied to a certain restricted class of locomotives, but is much in error under the normal operating conditions of freight engines especially. The relation between superheater and non-superheater locomotives

is open to criticism, and in the plot given in both of the sources quoted (note, p. 29) the tractive effort and horsepower curves are not mutually consistent.

The speed-factors developed from the University of Illinois tests give a curve correct in shape, but the ordinates are high in value as the engines tested were driven very hard. Very little may be said of the other speed factor curves, except that they are only worthy of consideration where the crudest estimate is required.

In other plots such as those of Figs. 3, 4, 5 and 6, the rational and semi-rational methods show a similar relation to actual performance, but the empirical methods show no such general relation. The starting point of each is fixed by the rated tractive effort, and the relation of the curves to each other is fixed, but the position of the actual test curve may either be in among them, as in the case of the average freight engine, or much higher at all but the lowest speeds, as would be expected in the case of the usual passenger engine with its much higher relative boiler capacity.

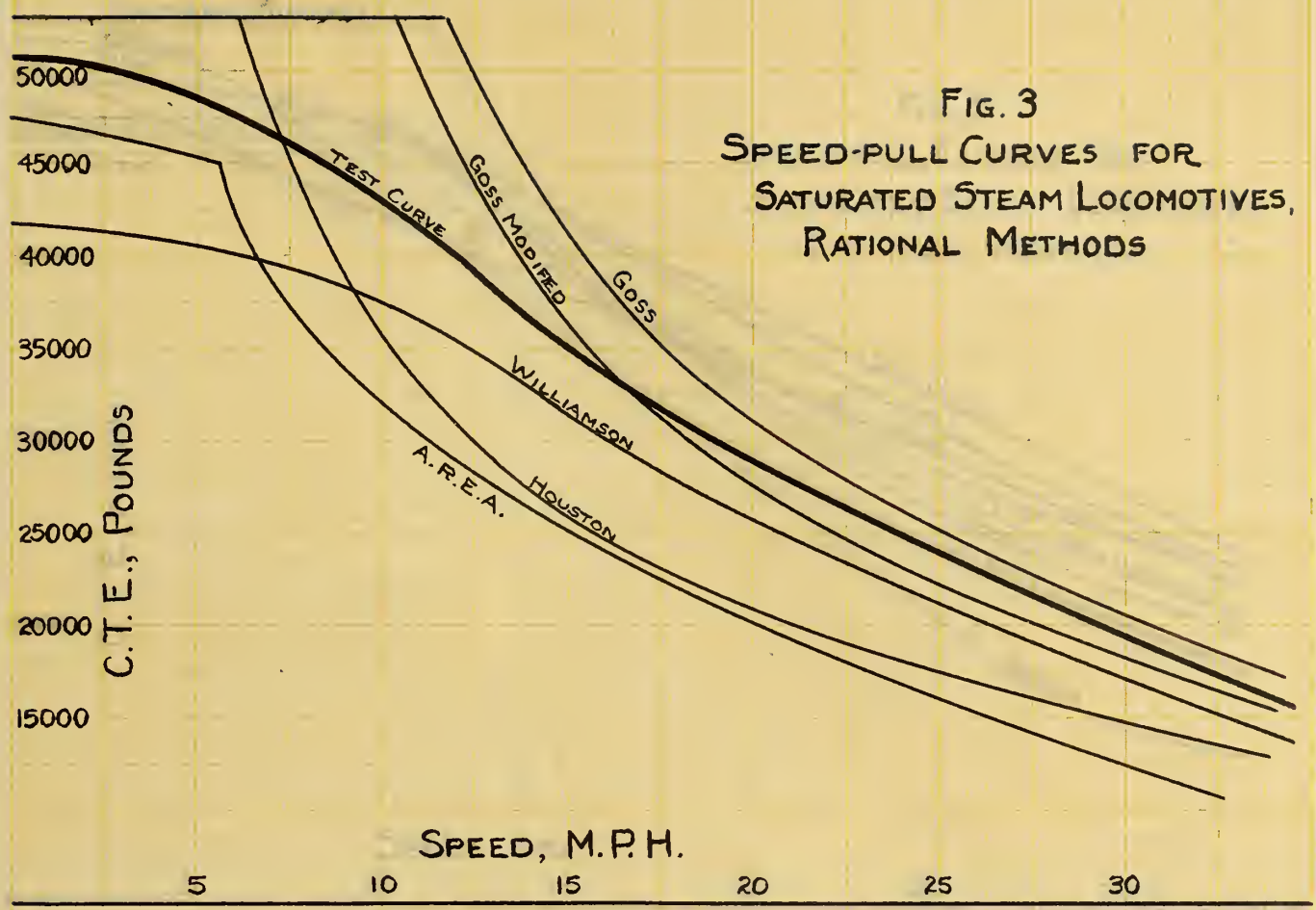

Fig. 3
Speed-pull Curves for Saturated Steam Locomotives, Rational Methods

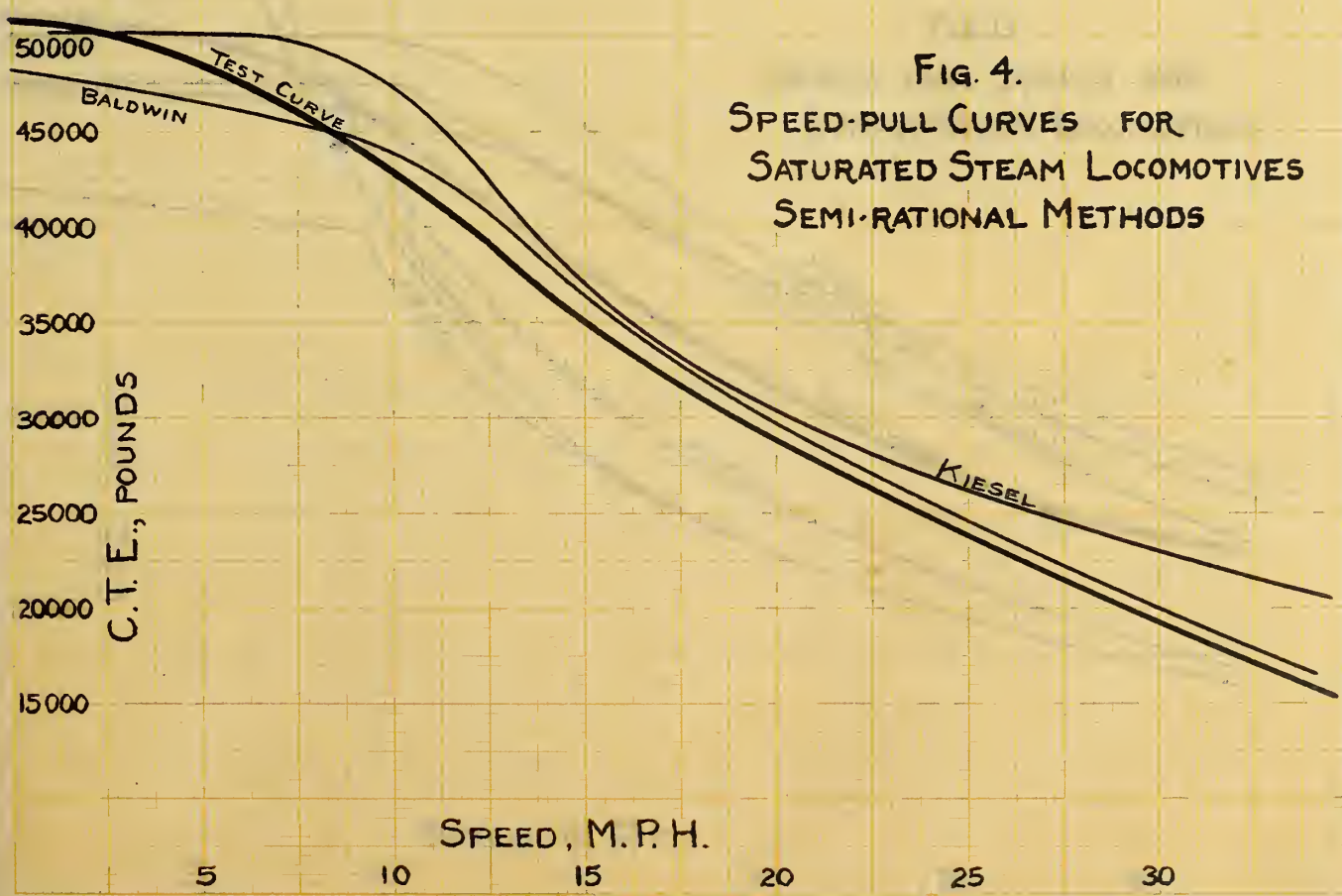

Fig. 4.
Speed-pull Curves for Saturated Steam Locomotives Semi-rational Methods

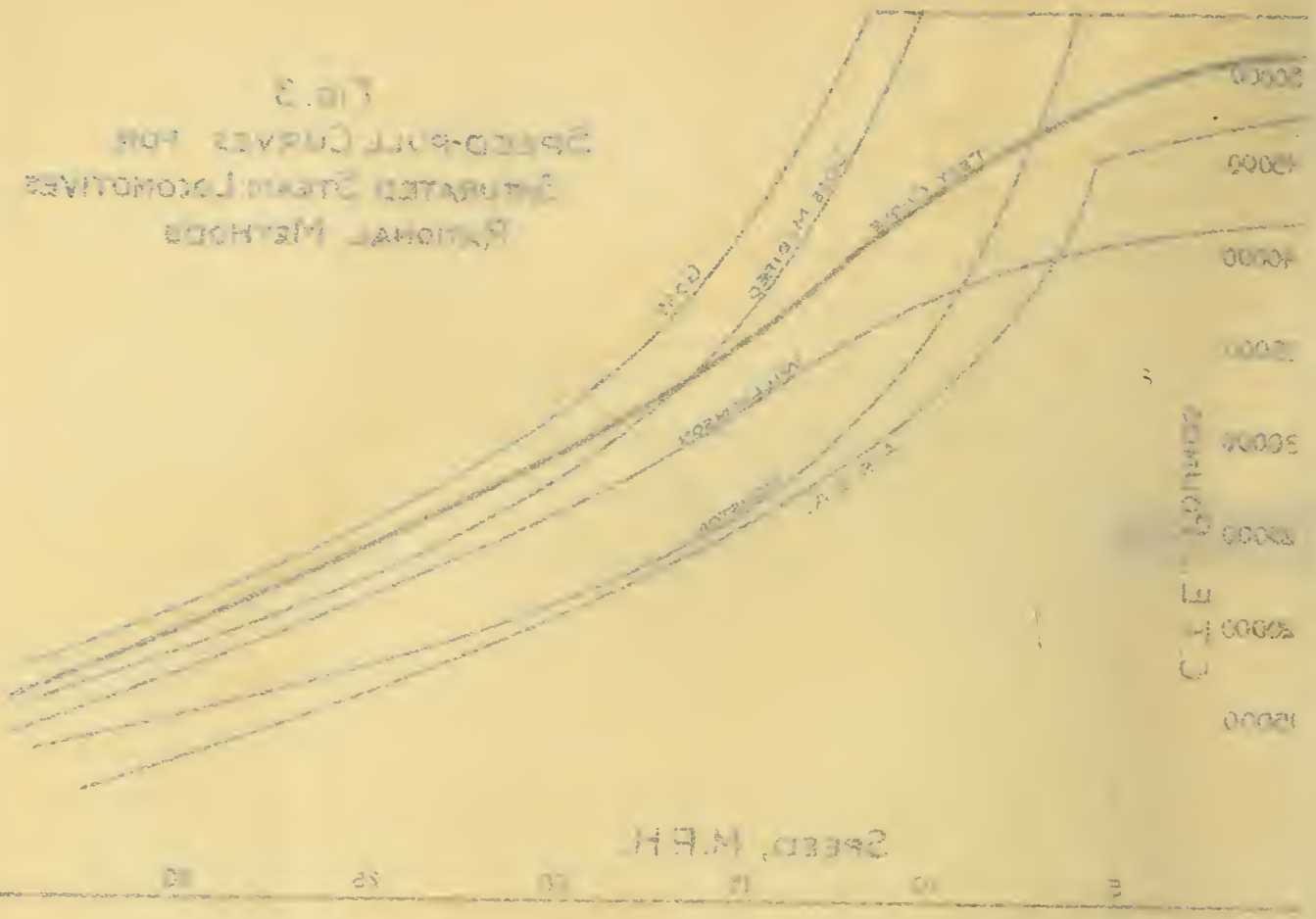

Fig. 3
Speed-Pull Curves for
Saturated Steam Locomotives
Rational Methods

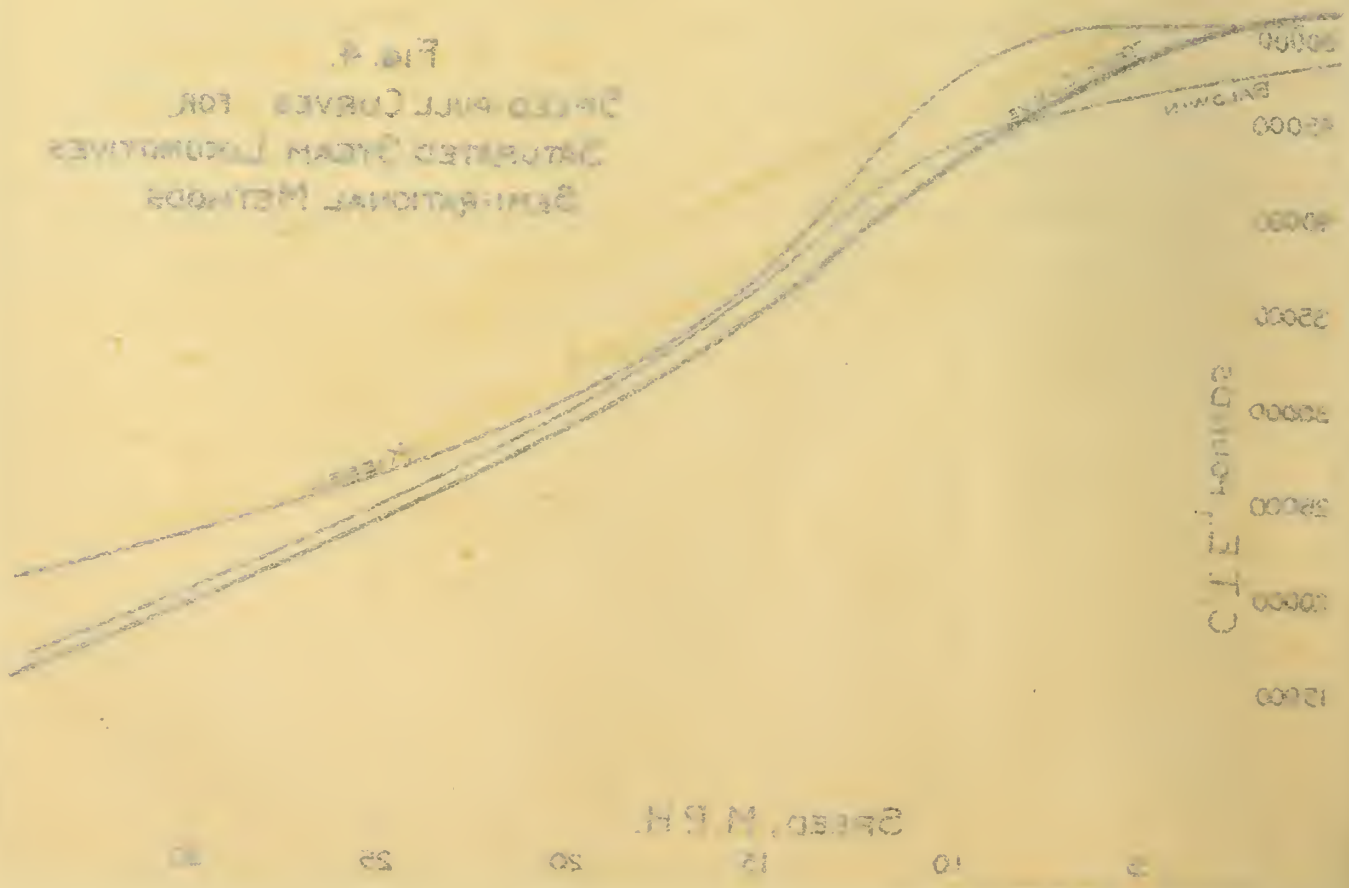

Fig. 4.
Speed-Pull Curves for
Saturated Steam Locomotives
Semi-Rational Methods

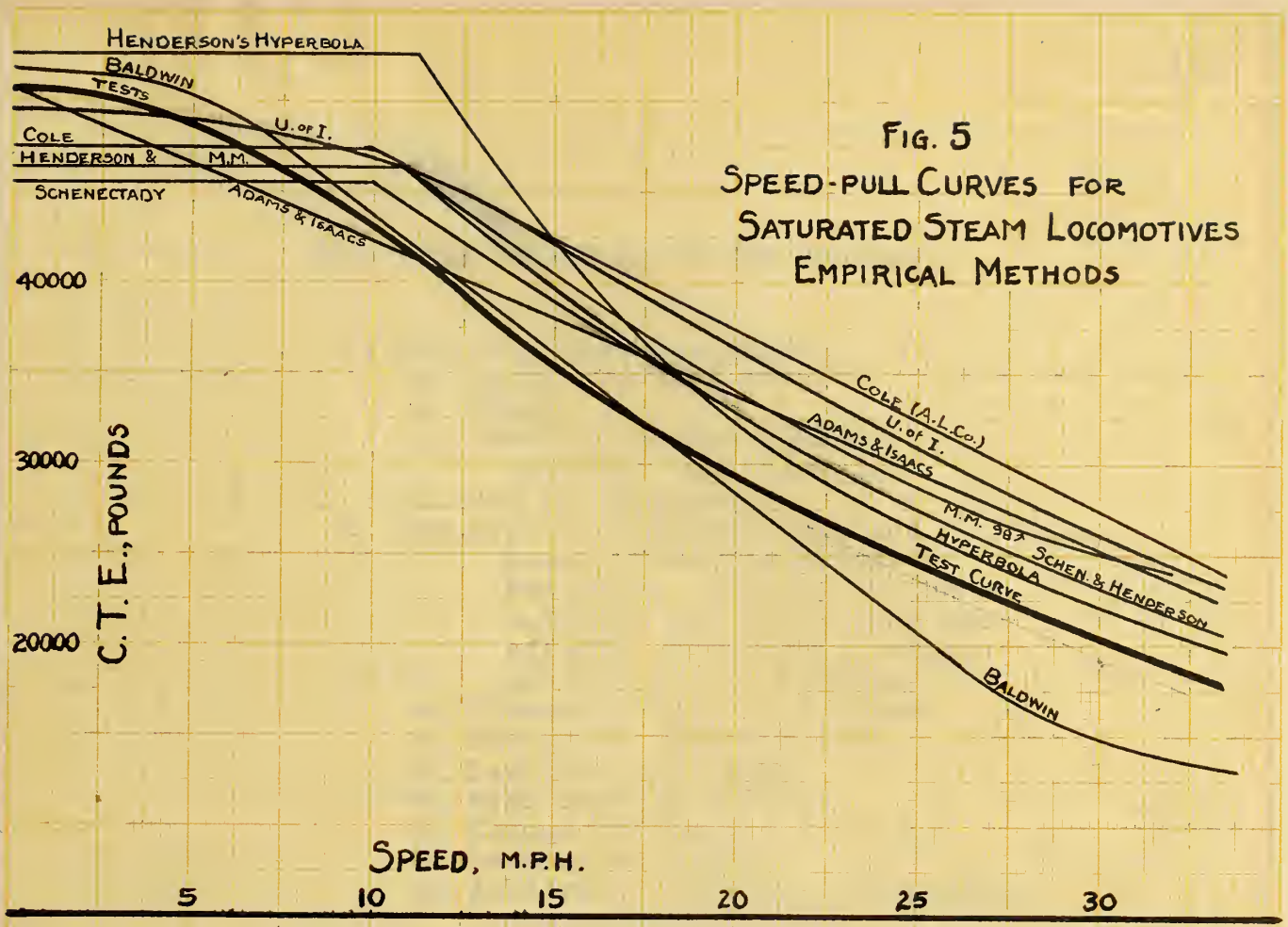

FIG. 5
SPEED-PULL CURVES FOR SATURATED STEAM LOCOMOTIVES EMPIRICAL METHODS

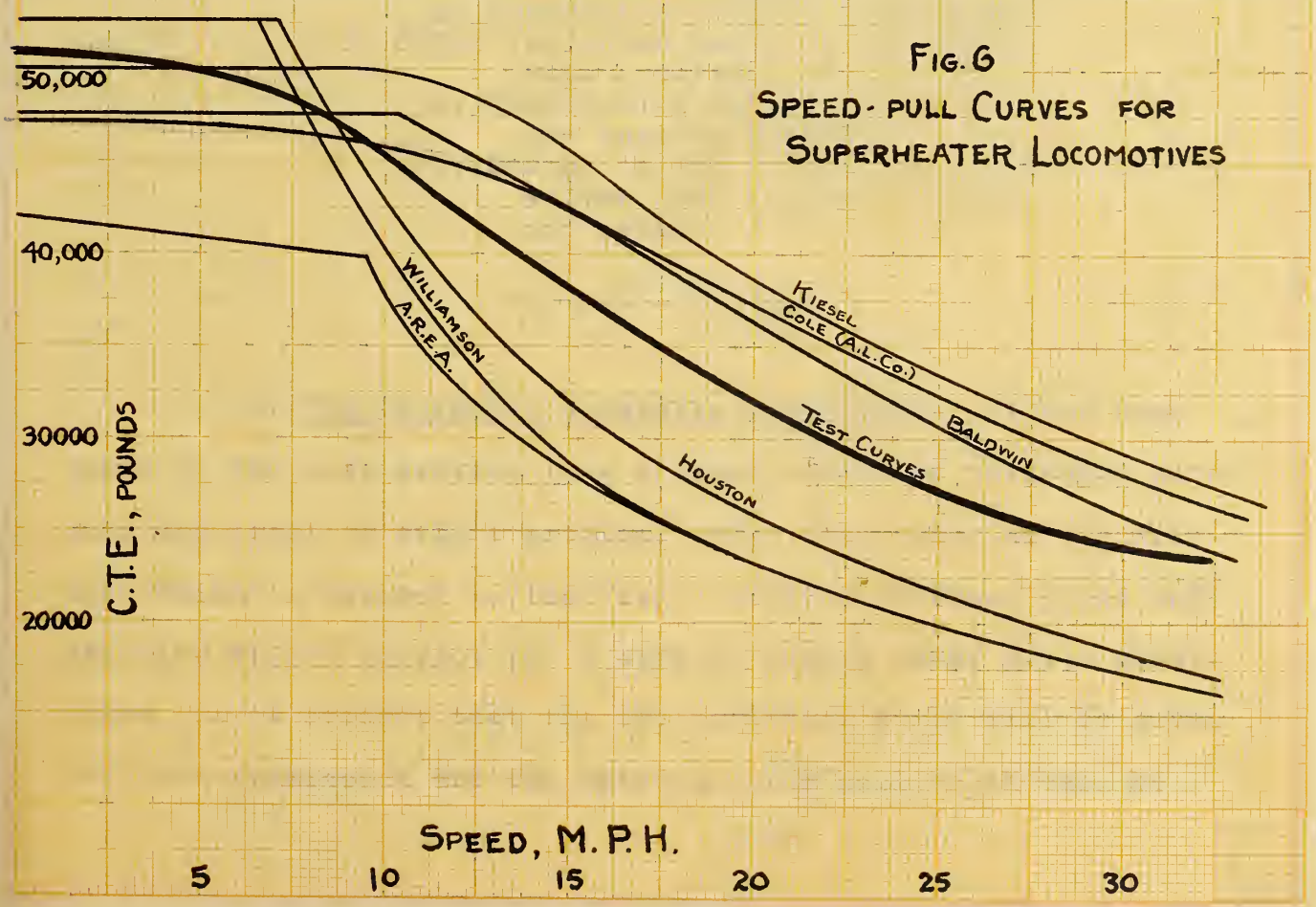

FIG. 6
SPEED-PULL CURVES FOR SUPERHEATER LOCOMOTIVES

III. THE VARIABLES OF THE PROBLEM.

1. The Principal Variables.
   a. Rated tractive effort.
   b. Speed.
   c. Heating surface (quantitatively considered).
   d. Use of superheat.
2. The Effect of these Variables upon a Tractive Effort Estimate: a Study of the A.R.E.A. method to show the effects of changing certain conditions, as follows:
   a. Change of heating surface.
   b. Change of cylinder size.
   c. Low rate of firing.
   d. High rate of firing.
   e. Change of quality of coal.
   f. Leakage of steam.
   g. Arbitrary increase of the evaporation estimate.
   h. Locomotive without superheater
3. Other Variables Entering the Speed-Pull Problem .
   a. Qualitative consideration of the heating surface.
   b. Design of the steam distribution: valves, valve-gears, piping, and ports.

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1. <u>The Variables Generally Considered.</u> It has been shown in the last section that a large number of variables enter into the tractive effort problem. From the nature of the relation which is desired -- that is, a relation between speed and tractive effort correct for a certain engine under given conditions, it is evident that the two variables which will be given the most prominence are the speed and the pull which can be

developed. The simplest statement of the speed-pull relation takes no account of any other variable than these two, and the weakness of such a method of estimation is evident. At the lowest speeds, that is, below six or eight miles per hour, it is perfectly proper to base a tractive effort estimate upon these two variables alone, but as soon as that speed is reached at which it first becomes necessary to reduce the cut-off in order not to decrease the boiler pressure, the number of primary variables becomes three. The third is the boiler capacity, and it very quickly becomes of greater weight than the cylinder size or rated tractive effort.

This statement may be put into concrete form by comparison with actual tests. Consider the cases of two locomotives tested by the Pennsylvania Railroad at Altoona, a Pacific and a Mikado. These engines had boilers of identical design and were fired at very nearly the same rate. At a piston speed of 800 feet per minute, the cylinder tractive force of the Pacific was 30000 pounds, and that of the Mikado, 36000 pounds, or twenty per cent greater. Any speed factor method would have estimated it to be thirty-eight per cent greater, this being the relation between their rated tractive efforts. Compare further two Atlantic type passenger engines, with equal rated tractive effort. At a piston speed of 1000 feet per minute, there is fifteen per cent difference in the pull developed, though any speed factor method would have credited them with the same tractive effort. This difference was simply due to the larger boiler capacity of one of the engines. These examples are sufficient to show that at high speed, boiler capacity has a larger effect than cylinder

size on the pull which can be delivered. This may also be shown in another way: For any speed, the limitations of steaming capacity prescribe a certain cut-off as the longest at which the engine may be operated without a decrease of boiler pressure, -- that is, the use of a greater amount of steam than the boiler is capable of supplying. The horse-power output of the engine varies directly with the steam available; the increase of the steam supply makes possible operation at a longer cut-off and a correspondingly higher tractive effort.

What is the effect, in the case of engines with the same steam supply available, of changing the cylinder size, the chief factor in rated tractive effort? A change in cylinder-size means a change in cut-off to utilize the same quantity of steam at the same speed; if the new cut-off is a more economical one, the horsepower-output will increase and with it the tractive effort in the same ratio. Extreme cut-offs, whether long or short, are uneconomical; the high water rate decreases the power output. It is possible therefore to <u>decrease</u> the tractive effort by <u>increasing</u> the size of the cylinders, and in some cases the opposite may also be true. However, under normal conditions, a small increase of cylinder diameter will improve the performance, as most engines seem to be under-cylindered for the best steam economy. Thus it is seen that at speeds above the critical point the rated tractive effort becomes of secondary importance, the effect of changing the cylinder-size having only a relatively small effect on the pull, due to the change in efficiency. On the other hand, it was shown that an increase in the boiler capacity affects the tractive effort in direct ratio.

Such a tractive effort method as that used by the Baldwin Works is a great improvement over the ordinary speed factor curve, since it in a way considers boiler capacity. Better results would be expected however from a similar set of curves actually based on evaporative capacity. The results given by the Baldwin curves, if correct for a certain rate of firing with a given grade of coal could not be correct for any other conditions unless the errors introduced should balance each other. In the Kiesel tractive effort formula, the simplest method of making an evaporation calculation is employed: the multiplication of the heating surface by an evaporation rate fixed by certain rules for estimating. No account is taken of the coal, as to its quantity or quality. Mr. Williamson assumes a rate of firing equal numerically to the heating surface area in pounds per hour and an average boiler efficiency, considering that the evaporation varies directly with the B.t.u. value of the fuel and with the heating surface. Mr. Houston adopts a rate of firing of 1-3/4 pounds of coal per hour per square foot of heating surface and produces from tests a table showing the evaporative value of coal of various heating values applicable to this rate of firing; when the rate is less, the evaporation is to be reduced in direct ratio. The A. R. E. A. method attacks the problem in a more logical manner, presenting a table showing the relation between the evaporation per pound of coal and the rate of firing per square foot of heating surface: in this way any quantity or quality of coal may be given due consideration. The assumption is made in this case also that the

evaporation varies directly with the heating value.*

The use or non-use of superheated steam must be considered as one of the primary variables, its effect showing in a greatly decreased water rate and consequently larger horsepower output. The only comparison that can be made between superheater and non-superheater engines is one between engines in other ways identical to show the advantage of superheat in any particular case. In general, the only way to take into account the superheat variable is by separating entirely superheater and saturated steam engines.

2. Effect of the Primary Variables in a Tractive Effort Estimate. Presenting the most logical method of estimating the evaporation, the A. R. E. A. method also proposes the most carefully thought out method of calculating tractive effort which has thus far been presented. The method is entirely rational, being based upon the estimated evaporation and the water-rate for various speeds, and the tractive effort is calculated directly from the horsepower. This method takes into consideration all of the three major variables and the results respond properly to almost any change in the conditions which may be made. In Fig. 7 are shown the results obtained from this

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* The evaporation study presented later will show clearly the fallacy of several of these assumptions: the evaporation does not vary directly with the heating surface, with the rate of firing, nor with the calorific value of the fuel. The characteristics of the boiler design have a much larger effect on the evaporation rate than the personal equation of the fireman, unless he be actually worked to the limit of endurance.

method plotted under nine different conditions -- these and a note on the results obtained in each case follow.

The normal curve is plotted for a superheater locomotive of the following dimensions and conditions:

Cylinders	24" x 30"	)
Drivers	60"	(pld^2/D = 57,600 pounds
Boiler Pressure ...	200 lb.	)

Heating Surface	3244 sq. ft.
Grate Area	65 sq. ft.
Superheating Surface ..	700 sq. ft.
Tubes	280, 15.5' long; diameter 2"
Superheater flues	36, 5-3/8" diameter
Coal used	4000 lb. per hour
Thermal Value	13,000 B.t.u.

The first modification is to show the effect of a slight increase in the heating surface, the new surface being 4400 square feet. The evaporation is increased, and hence M is increased: the increase in horsepower output is not as great as the increase in M, however, which results in a slightly lower value for the tractive effort at M. Beyond this point the curve is higher at all points than the standard.

The second modification is an increase of cylinder size to 26" x 30". The critical speed occurs slightly earlier, and the tractive effort is higher than the adhesion limit. As the speed increases the curves become coincident.

The third modification is in the use of a smaller amount of coal, the firing rate assumed being 3000 pounds of coal per hour. The speed M is lower, and the tractive effort at M is also lower, as is the curve throughout.

The effect of a high rate of firing, 5000 pounds per hour, is also shown, the critical speed being higher, also the

tractive effort at all points. The result of firing 4000 pounds, (the standard amount) of coal of 15000 B.t.u. quality gives almost identical results.

Actual measurements* have shown that a leak of almost 3000 pounds per hour is a possibility in each valve. In order to represent the effect of some such waste of steam, whether by leaks or other drains on the boiler the curve marked "leak" is plotted, 4000 pounds of steam having been subtracted from the normal evaporation.

The next variation considered is that of the evaporation as a whole. The performance of similar boilers and every other method of estimating the output of a boiler all tend to show that the A. R. E. A. estimate is very conservative. In Appendix III-B are given the evaporation per hour obtained by various estimates, and these may be summarized as follows:

A.R.E.A., as used for normal conditions, 4000 pounds of 13000 B.t.u. coal per hour, 23,400 pounds per hour.

Cole (A. L. Co.) method, taking no account of coal fired, 42,600 pounds. Applying "coal factor" for 1300 B.t.u., 35,500 pounds.

Houston's estimate, 27,800 pounds.

Williamson's estimate, 35,500 pounds.

Method of Section VII, 26,700 pounds.

The tractive effort of this engine has been calculated on the basis of the highest estimate, or 35,000 pounds evaporated per hour. The speed M is high, and being relatively higher than

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* Proc. Master Mechanics' Association, Vol. XXXVII, p. 274.

the increase in horsepower, the tractive effort at M is lower than the normal curve.

The final variation shown is the same engine without the superheater. A slightly larger evaporation and a much lower value of the speed M do not counteract the effect of the high water rate, with the result that the tractive effort at M is low. Between ten and twenty miles per hour the curve lies close to the lowest superheater curve, and at higher speeds falls off faster and lies below all of the others.

It will be seen that the value of the tractive effort at M varies from 50,300 pounds in the normal curve to 48,500 in the curve representing the high evaporation estimate, (excluding the curve for 26-inch cylinders) due to the varying relation between the speed and the horsepower at M. No reason appears for any variation in the value of the tractive effort at this point except in the case of the leaking engine, where a lower pull would be expected in case the leak were in the cylinders. Neither does any reason appear for a variation in the starting tractive effort such as is shown. The method would be strengthened by relating it more closely to the cylinder tractive effort, and this could be done by assuming that a certain percentage of the theoretical tractive effort, say 95%, can be maintained to a point representing half of the speed M, then round the corner off connecting the straight line with the curve through the range of speed from .5 M to M. Where the point M is above the adhesion limit, or more properly above the pld^2/D limit, the 95% line should be used, extending it to the point where the curve

is intersected, rather than making the adhesion limit the basis for the maximum pull as is now done.

3. Other Variables. Where the heating surface has been used in a study of engine output, with two exceptions* the consideration has been entirely a quantitative one, no weight having been given to the arrangement or disposition of the area, or the effect of these elements upon the evaporative efficiency. Since it is an estimate of the evaporation that is sought, such methods leave much to be desired. The relation existing between the grate area and the tube-heating surface, and the length of the tubes are the principal features of the design which cause wide ranges in the efficiency. Since an accurate evaporation estimate is essential to the production of a dependable speed-pull relation, the variation in boiler efficiency should be considered one of the main variables -- or the evaporative capacity should be substituted for the heating surface.

One variable of considerable importance remains, and this is the design and condition of the steam distribution system, the efficiency of which has a large influence upon the water-rate, and may become the controlling factor. For any given speed and cut-off the following factors influence the water rate:

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* The exceptions referred to are the Cole method, which evaluates the efficiency of the disposition of the heating surface, but ignores the quantity and quality of the coal fired, and the first method proposed by Mr. Shurtleff, previously referred to. Mr. Henderson makes a study of efficiency, in "Locomotive Operation", but does not use it in estimating tractive effort. These methods are further considered in Section VII.

The use of superheated steam or saturated steam, and to a minor extent the degree of superheat.

The very small influence* of the boiler pressure.

The influence of the steam distribution.

This last has never been taken into consideration in any method of estimating tractive effort. It is probable that the very uniform trend and the similar values of water-rate curves for superheater engines are due in a large measure to the almost identical design of steam distribution parts now in use, and the wide variation in the same curves for saturated steam locomotives is due in part to exactly the opposite conditions.

The steam distribution system consists of four essential parts: the steam-pipes, the ports, the valves, and the valve-gear. The advantage of pipes with long radius bends and generous diameter is obvious, and the same may be said of generous port-openings, due to the reduction in friction effected. Many tests have proved that the double-ported valve is more efficient than the plain D type, and the piston valve more efficient than either. As to valve-gears, no inherent advantage making for efficiency in steam consumption is apparent, but the fact that outside gears are more easily adjusted, and remain in better adjustment is a factor that cannot be overlooked when service conditions are being considered. Saturated steam engines may be roughly divided into three classes then, for comparison:

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* The Purdue tests reported in Proc. Master Mechanics' Ass'n. (Vol. 42, 1909, p. 144) show that 200 pounds is the most economical pressure for superheater engines, and for saturated steam the most efficient pressure is higher than any now in use.

Class A: Locomotives having the improved steam-distribution characteristic of superheater locomotives; piston valves and the resulting generous ports, outside steam pipes, and outside valve gears.

Class B: Locomotives having piston valves only.

Class C: Locomotives with slide-valves.

The performance of these three classes will be studied separately in an effort to discover the effect of the steam distribution variable.

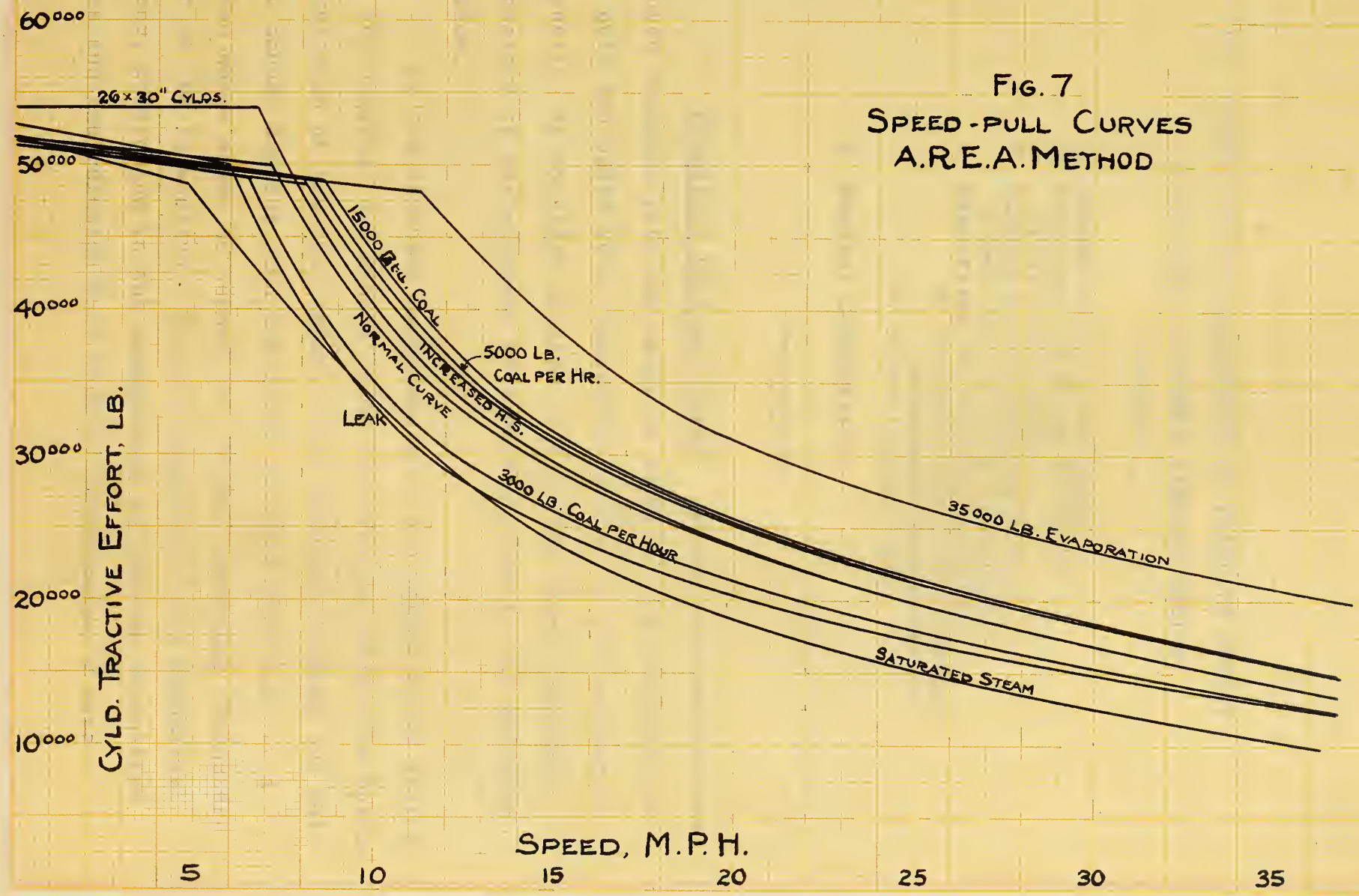
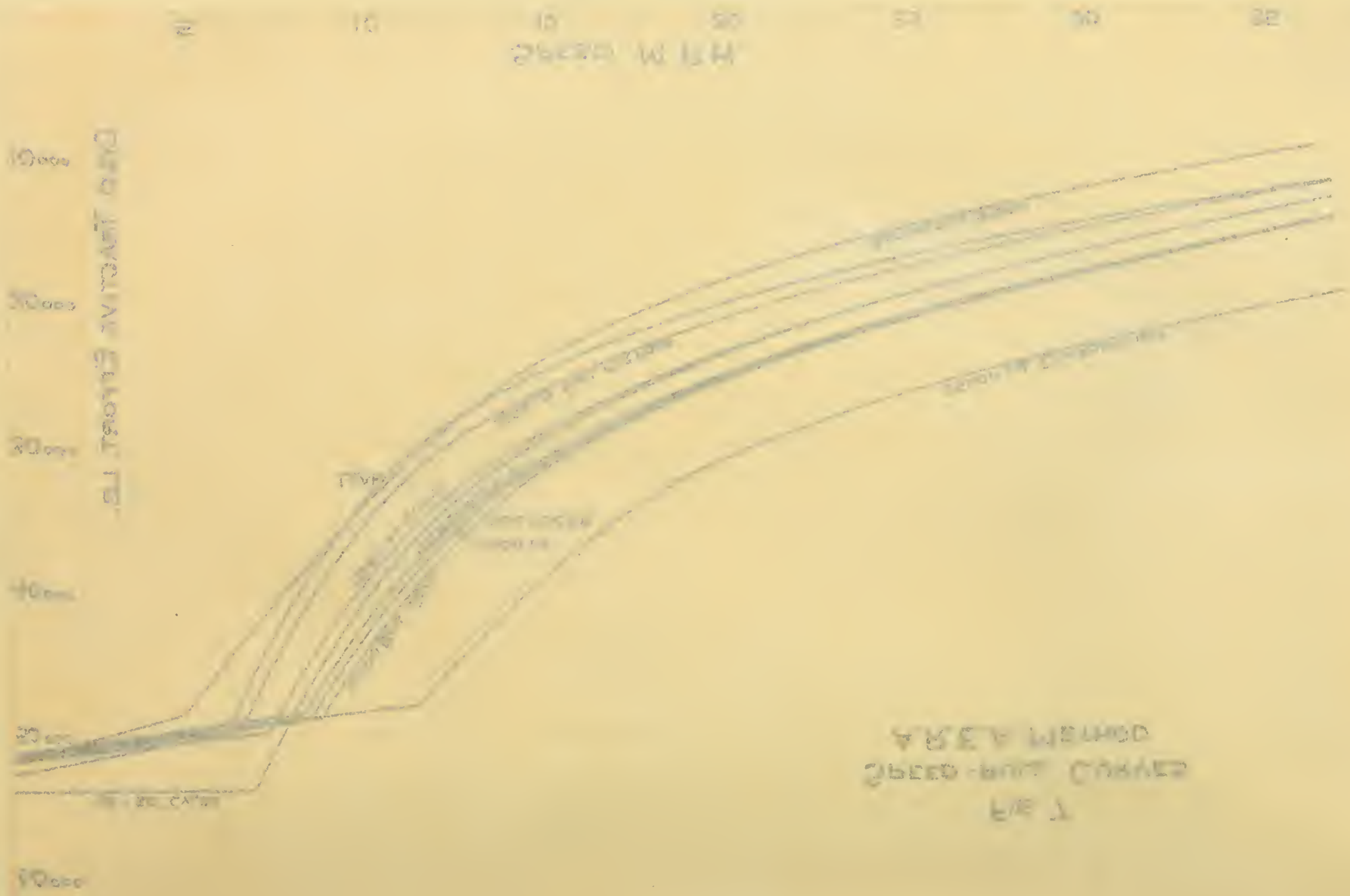


FIG. 7
SPEED-PULL CURVES
A.R.E.A. METHOD



IV. BASES FOR THE COMPARISON OF TRACTIVE EFFORT IN ENGINES OF WIDELY VARYING DESIGN.

1. Pounds of Pull and Speed.
2. Fractions of Rated Tractive Effort and Speed.
3. Elimination of Superheat and Steam Distribution Variables.
4. Comparison on Evaporation Basis.
   a. Indirect comparison through horsepower calculations.
   b. Direct comparison.
5. General Comparisons.

1. Pounds of Pull and Speed. In the last section the variables entering into any complete study of the relation between pull and speed were enumerated, and it is the purpose of this section to consider methods of relating these variables, introductory to the analysis of specific cases in the sections to follow.

To the official concerned with the operation of trains, or to the engineer studying a grade reduction problem, the speed-pull relation of the most value is the relation between the pull at the tender drawbar, and the speed in miles per hour, -- a relation which might be termed the "normal operating characteristic" of the locomotive. For the purposes of the mechanical engineer, interested in the comparisons of various locomotive designs and the relation they bear to performance, a more

satisfactory relation is based upon cylinder tractive effort and indicated horsepower. These should be substituted for those produced at the drawbar if an accurate study is to be made.

As to the speed, it is evident that the mean effective pressure falls off due to the speed with which the piston travels and since the piston speed bears a relation to the speed of the engine which varies widely, for a fair comparison of engines of different design, the speed used should be that of the piston. We need, then, a relation between cylinder tractive effort and piston speed, and if several engines are represented, certain conclusions may be drawn. Those engines having the highest rated tractive efforts have the highest pulls at low speeds -- an obvious fact. Also, of two engines having rated tractive effort nearly equal, the one with the larger boiler capacity has a higher pull at high speeds, due to the fact that more steam is available.

2. Fractions of Rated Tractive Effort and Speed. Given a set of curves such as referred to in the preceding paragraph (or those in Fig. 10) showing the relation between pull and speed for several engines, there would be no method of estimating the relation which should exist for a comparable locomotive. Conversely, knowing the relation, desired, there is no method of designing a locomotive to produce the curve -- there are too many variables remaining. Since the purposes suggested are the objects of such a study, the investigation must be carried further in order to eliminate some of the variables. In the previous investigations of the subject the step generally taken

has been the removal of the rated tractive effort factor by plotting the relation for each engine in terms of the percentages of the rated tractive effort which could be developed at the various speeds. In such a set of curves, all of the zero-speed points should lie very close together, some point between 90 and 100% of the value of pld^2/D representing the average.

3. Elimination of the Distribution and Superheat Variable. The effect of variation in the theoretical tractive effort being eliminated, the variables remaining, in addition to the percentage of pld^2/D and the speed, are those of boiler capacity, steam distribution, and the use or non-use of superheat. Of these three, two more may be removed by another method; a separate treatment by classes. This is essential in the case of the use of superheated steam, there being no basis for a fair comparison between superheater and non-superheater engines for technical purposes except the comparison of engines otherwise identical. In the case of the steam distribution, separation by classes is much less essential, but this separation furnishes a method of investigating the effect of steam distribution design on the speed-pull relation. We have, then, using the steam-distribution classification given in the last section, six classes of engines theoretically, each of which must be separately studied. These are:

1.	Superheated Steam,	Class A.
2.	" "	Class B.
3.	" "	Class C.
4.	Saturated Steam,	Class A.
5.	" "	Class B.
6.	" "	Class C.

Classes 2 and 3 are virtually non-existent. The Purdue locomotive Schenectady No. 3 is of class 3, but no other data from similar engines are available to compare with it. It is probable that a few engines of Class 2 have been produced by the conversion of saturated steam locomotives to superheater engines, but no test data are available. These two classes may be dismissed as far as further detailed study is concerned, leaving the performance of four classes to be studied.

Now if a series of curves for the speed-pull relation of several engines in the same class be plotted, and the data be used as previously stated (percentage of rated tractive effort and piston speed) only one variable remains to influence the forms of the curves given by different engines -- the boiler capacity. This set of curves should be identical in general form to the Baldwin speed factor curves (See Fig. 8), and if sufficient data were available it is doubtless true that such a showing could be produced. According to what has been previously stated, the different forms of the curves should be credited rather to boiler capacity than to mere heating surface, as in the case of the Baldwin curves. In such a plot, it would be possible, knowing the boiler capacity for each engine represented, to interpolate the curve for a comparable engine, or reversing the conditions, to determine the dimensions of the engine from the curve which is required, assuming that the rated tractive

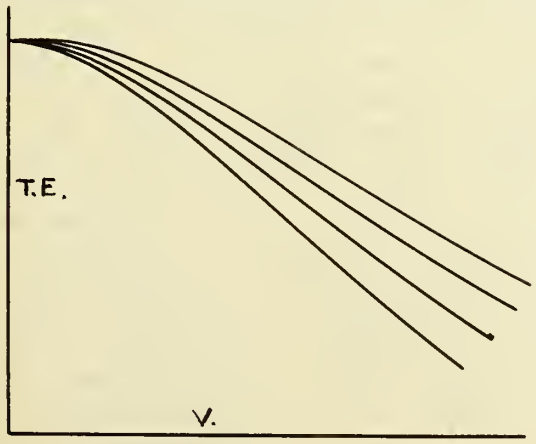


Fig. 8. Speed-Pull Curves based on Tractive Effort.

effort requirements are already known from other sources.

The methods used in the rational tractive effort estimates previously discussed suggest another basis for comparing the tractive effort and of giving the boiler capacity variable even greater prominence. This is by basing the comparison upon the evaporation rather than on the rated tractive effort. All of the methods investigated figured the tractive effort through the medium of the indicated horsepower -- dividing the evaporation estimate by the water-rate estimate to obtain the power, and applying the formula T = 375 HP/V to obtain the tractive effort. The transition from evaporation to tractive effort may be made without the interposition of the power, however. This is done by expressing the tractive effort as a percentage of the hourly evaporation, or in terms of "pounds pull per pound of hourly evaporation".*

The two comparisons may be made clearer by an illus-

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* A physical interpretation of this new unit of "pounds pull per pound of hourly evaporation" may be reached through the familiar curves of engine performance. One of these is the curve between water-rate and indicated horsepower: this may be inverted and interpreted for any point "the number of horsepower per pound of steam at the given horsepower". A given horsepower implies a certain speed, so the interpretation may be altered to "the number of horsepower produced by a pound of steam at a given speed". We would then have a curve between horsepower (per pound of steam) and speed, from which a curve of tractive effort per pound of steam and speed can be directly derived mathematically. Any point on the curve may be interpreted as the number of tractive-pound-hours which can be produced by a pound of steam at the given speed. Tractive-pound-hours, coupled with speed is a measure of work; pounds of steam is a measure of heat; the ratio then is efficiency -- or an implied water-rate. The assumption is made that the relation between piston speed and water rate is the same for engines in the same class, as given on p. 62.

tration. Suppose an experimental locomotive could be built with the boiler partitioned off in such a way that various fractions of the total capacity might be developed. The operation of this engine with varying amounts of boiler output, the rated tractive effort being constant, represents the comparison between the curves for the percentage of rated tractive effort and piston speed for several engines. Assume another experimental engine of fixed boiler capacity and variable cylinder size: the curves representing the speed-pull relation for this locomotive with various rated tractive efforts and the same boiler capacity represent the second comparison, where the evaporation is the base. These conditions are actually met in service in a familiar way: the boiler capacity and hence the tractive effort may be varied at all but the lowest speeds by the firing of more or less coal; the operation of a locomotive at less than the maximum cut-off which can be maintained at the operating speed is equivalent to a decrease in cylinder size.

The geometrical relation between each of the curves of the last plot -- that is, between piston speed and pounds pull per pound of evaporation -- is fixed by the relation between the tractive effort rating of the engine and evaporative capacity of the boiler. If such a series of curves show by their concordance that the assumption of equal water rates in comparable engines is justified, a curve may be interpolated with certainty to represent the performance of another engine: it remains to test the data for their concordance, and if

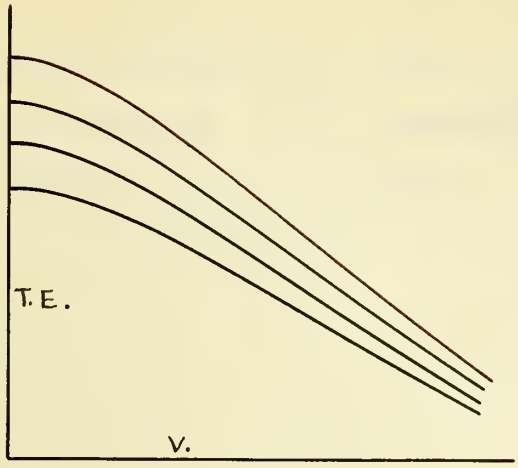

Fig. 9.
Speed-pull Curves Based on Evaporative Capacity.

satisfactory, to produce a correct method of interpolation. If this can be done for each of the four classes of engines, a method of estimating tractive effort will be produced which takes into account all of the variables -- rated tractive effort, speed, boiler capacity, steam distribution, and the use or non-use of superheat.

In working out the design for a new engine, a method of estimating the evaporation is required -- this is developed in Section VII, taking into account the design of the boiler, the quantity and quality of the fuel fired. It will be noted that several previously used assumptions are avoided, such as:

A single firing rate applicable to all engines.
A fixed firing rate for a given engine.
The evaporative value of all coals the same.
A fixed water rate for all engines (i.e. all saturated engines)
A fixed variation between water rate and speed for all engines using saturated steam.
A constant boiler efficiency.
An average relation between boiler capacity and rated tractive effort.
That boiler capacity varies directly with total heating surface, total coal fired, or B.t.u. value of the coal.
That the curve produced represents "all-day performance" or the opposite, the engine "working to the limit".

Finally it is worthy of note that in a semi-rational method such as proposed, based upon actual test curves, there is no uncertainty concerning the value of the tractive effort for speeds

around the critical point -- the importance of this fact being self-evident in the consideration of freight engines and their capacity on ruling grades.

V. TRACTIVE EFFORT VARIATION IN SUPERHEATER LOCOMOTIVES.

1. Comparison of Results of Laboratory Tests.
   a. Pounds pull and piston speed
   b. Fraction of rated tractive effort and piston speed
   c. Pounds pull per pound of evaporation and piston speed
2. Development of the Relation between Boiler Capacity and the Pull at 600 ft. per minute Piston Speed.
3. Construction of Curves for Other Speeds.

1. Comparison of Results of Laboratory Tests. The most reliable means of making a study of any phase of locomotive performance in which the capacity of the boiler is an important factor is through test data obtained from laboratory trials. In a road test, the exact amount of steam delivered to the cylinders is unknown on account of the amount consumed by the auxiliaries. Further, only an average rate of evaporation can be obtained over an entire test, through which the locomotive may have been operated at widely varying rates of power-output: this is particularly true in freight-service tests, where stops may be frequent and long. Since tractive effort is so closely related to boiler output, testing-plant records provide the best basis for a study of the relation existing between pull and speed,

and there are now available a sufficient number of laboratory tests of locomotives using highly superheated steam to warrant efforts in the direction of reconciling and generalizing the results. From the tests made on the Altoona plant of the Pennsylvania Railroad, data have been published for the following engines using superheated steam: three passenger engines of the Atlantic type, three of the Pacific type, and two freight engines -- one Consolidation and one Mikado.

The curves showing the speed-pull relation in every case are plotted on a basis of speed in miles per hour, and drawbar pull. While this form of comparison is the most valuable to the official of the operating department, it is such that no general relation can be discerned between the different curves. The following facts are clearly shown, however. A high rated tractive effort produces a large actual pull at low speeds -- an obvious fact. When the rated tractive efforts are equal, the engine with the larger boiler capacity has the larger pull at high speeds, an almost equally obvious fact; since for a given speed, tractive effort varies directly with the horsepower, and horsepower with the evaporative capacity. For purposes of an accurate comparison, however, certain modifications should be made in these curves. Since it is the difficulty of filling the cylinder at high speed which is in the main responsible for the lower pull, the speed of the piston is a better measure of the performance than that of a point on the rim of the drivers. For example, if an engine with eighty-four-inch drivers and a twenty-four-inch stroke travels thirty miles an hour, the average speed of the piston would be 480 feet per minute; an engine

with fifty-one-inch drivers and a thirty-inch stroke at the same speed would have a piston-speed of 990 feet per minute. It is plain that it would be easier to fill the cylinder in the former case, though the speed in miles per hour is the same. Also the original curves make no allowance for friction and this must be eliminated, otherwise the four-driver engines have an advantage over those with three and four pairs. In the publications of the Pennsylvania Test Department the friction losses for all of these engines may be found.* The original curves, reduced to a basis of cylinder tractive effort** and piston speed are shown

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* See Bulletins 10. 11, 18, 19, 21, 22, 26, 27, and 29.

** By "cylinder tractive effort" is meant that pull which could be developed if the moving parts were frictionless. It may also be defined mathematically as the pull which multiplied by the speed of the engine and divided by the appropriate constant will give the indicated horsepower.

In Figs. 10, 11 and 12, the solid lines represent the range of pulls actually obtained on the testing plant. The dotted lines represent pulls that have been estimated in various ways to complete the curves. The estimations have been made as follows:

The very low speed sections of the curves for the E6s (new), the H8sb and the L1s, from 1 to 100 feet per minute piston speed are completed on the assumption that 98% of the boiler pressure may be developed at starting, an assumption well supported by the pulls obtained at the lowest speeds.

The curves of the other engines are completed to zero speed on the same assumption, combined with the calculated curves given in the test reports.

The curves for the two freight engines are completed through the high-speed range by plotting the horsepower curves, and from them deriving the tractive effort curves. The horsepower curves show that at the highest test-speeds the L1s and the H8sb are slightly below probable maximum horsepower output. (See Fig. A1, Appendix V-C).

The curve for the K2as is completed from 1200 to 1400 feet per minute piston speed on a basis of constant horse power, as the test at the highest speed was below the capacity of the boiler. The estimates mentioned in the first two paragraphs are not made use of in any way in the discussion which follows.

In Appendix V-A will be found the dimensions for the engines, and in Appendix V-B the data for figs. 10, 11 and 12.

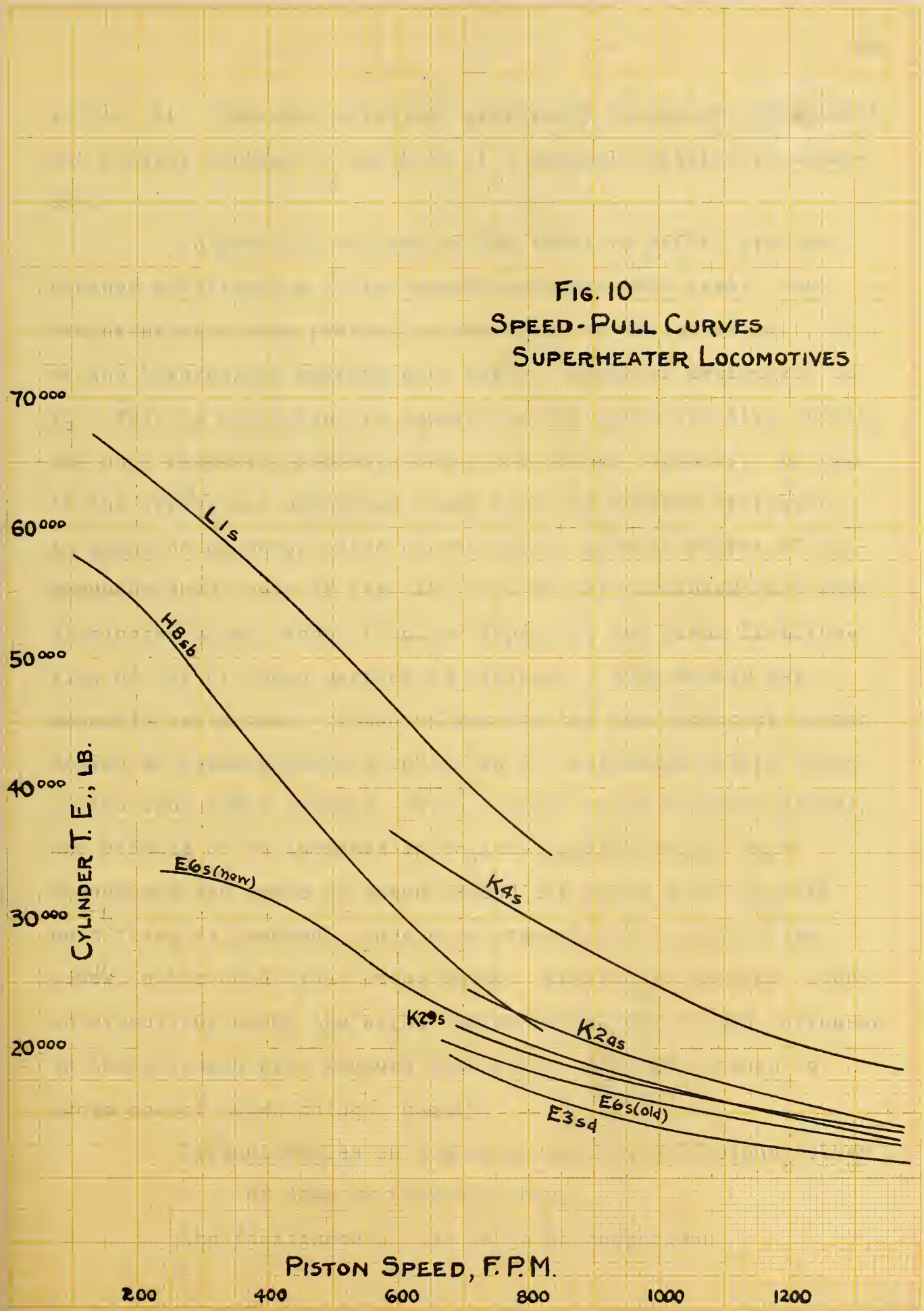
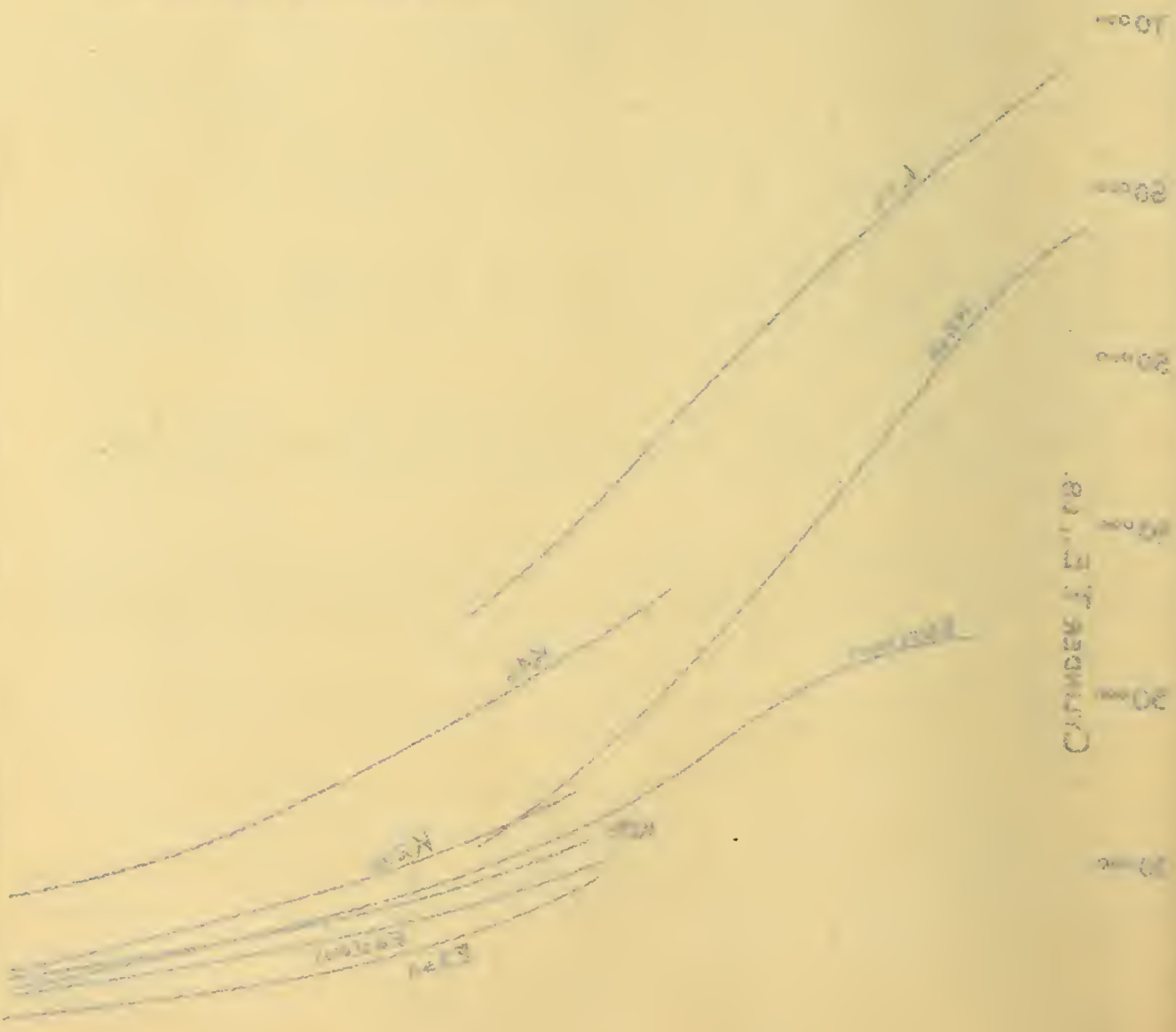


FIG. 10
SPEED-PULL CURVES
SUPERHEATER LOCOMOTIVES
70000
60000
50000
40000
30000
20000
CYLINDER T. E., LB.
L1s
H8sb
E6s(new)
K4s
K29s
K2as
E6s(old)
E3sd
PISTON SPEED, F.P.M.
200
400
600
800
1000
1200



in Fig. 10. Here the relations previously suggested are evident, but nothing further in the line of a general relation is apparent.

In previous studies of the tractive effort problem another modification of the coordinates has been used: the ordinates have been plotted as fractions of the rated pull, or of the theoretical maximum pull pld^2/D, hereafter designated as T_o. This is equivalent to equalizing the rated tractive effort, the only remaining variable being the boiler capacity. In Fig. 11 the curves are presented again with the altered ordinates. As might be expected these curves show a greater degree of concordance than those of Fig. 10. All of the variables have been eliminated except that of boiler capacity; the steam distribution of all of these engines is similar -- they embody Walschaerts valve-gear, piston valves and the best practice in the design of steam-pipes and ports, so no difference should be expected from these sources. With a given rated tractive effort, the effects of an increase in boiler capacity should show throughout the range of speed beyond the point where cut-off must first be reduced, since more steam produces more horsepower, other conditions being equal. There are, however, eight intersections among the eight curves shown, and slight increases in their length will produce five more. This fact tends to prove one of three things, namely:

Irregularities in the data, and not sufficient volume of data to overcome them.

The faultiness of the basis of comparison.

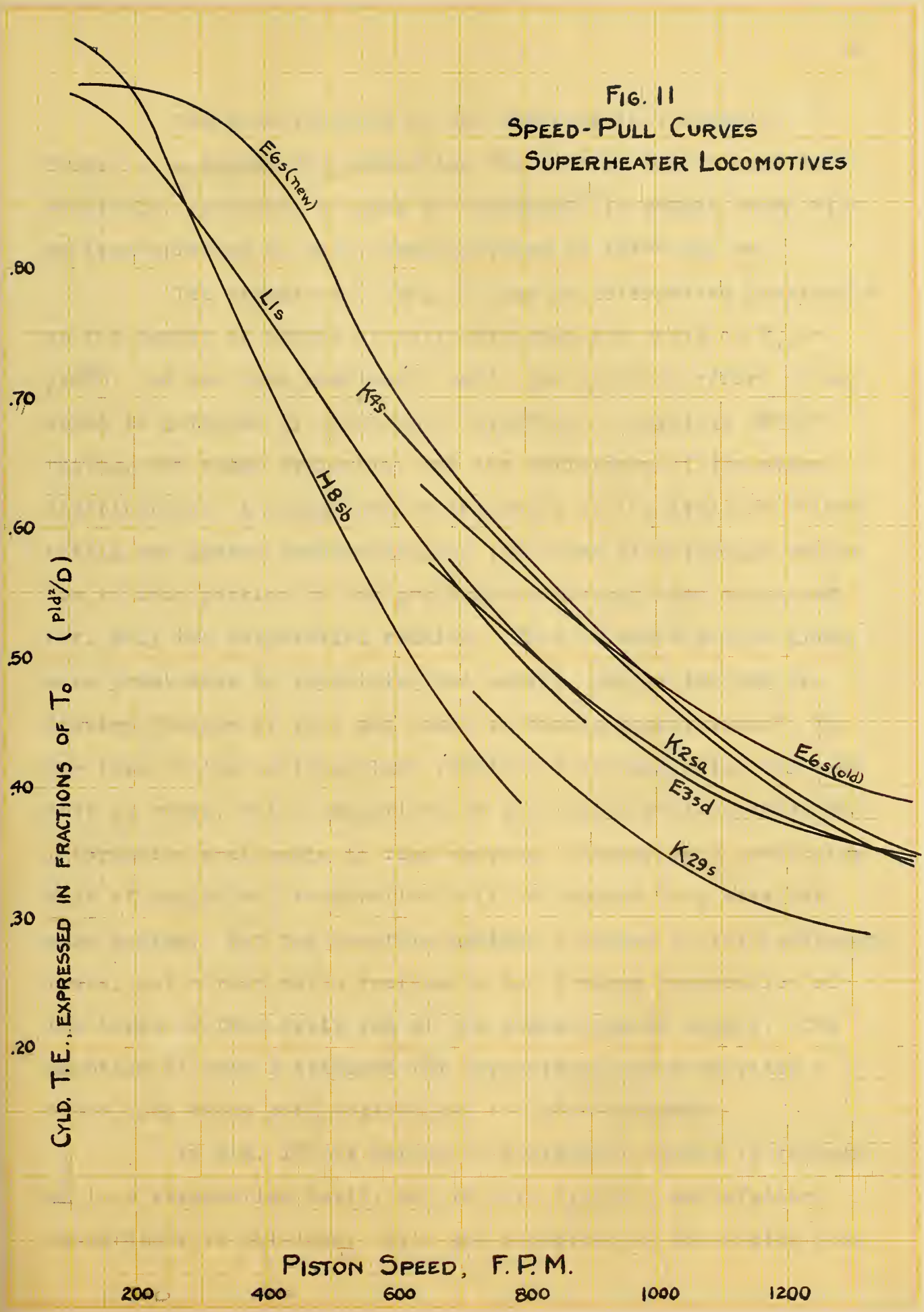
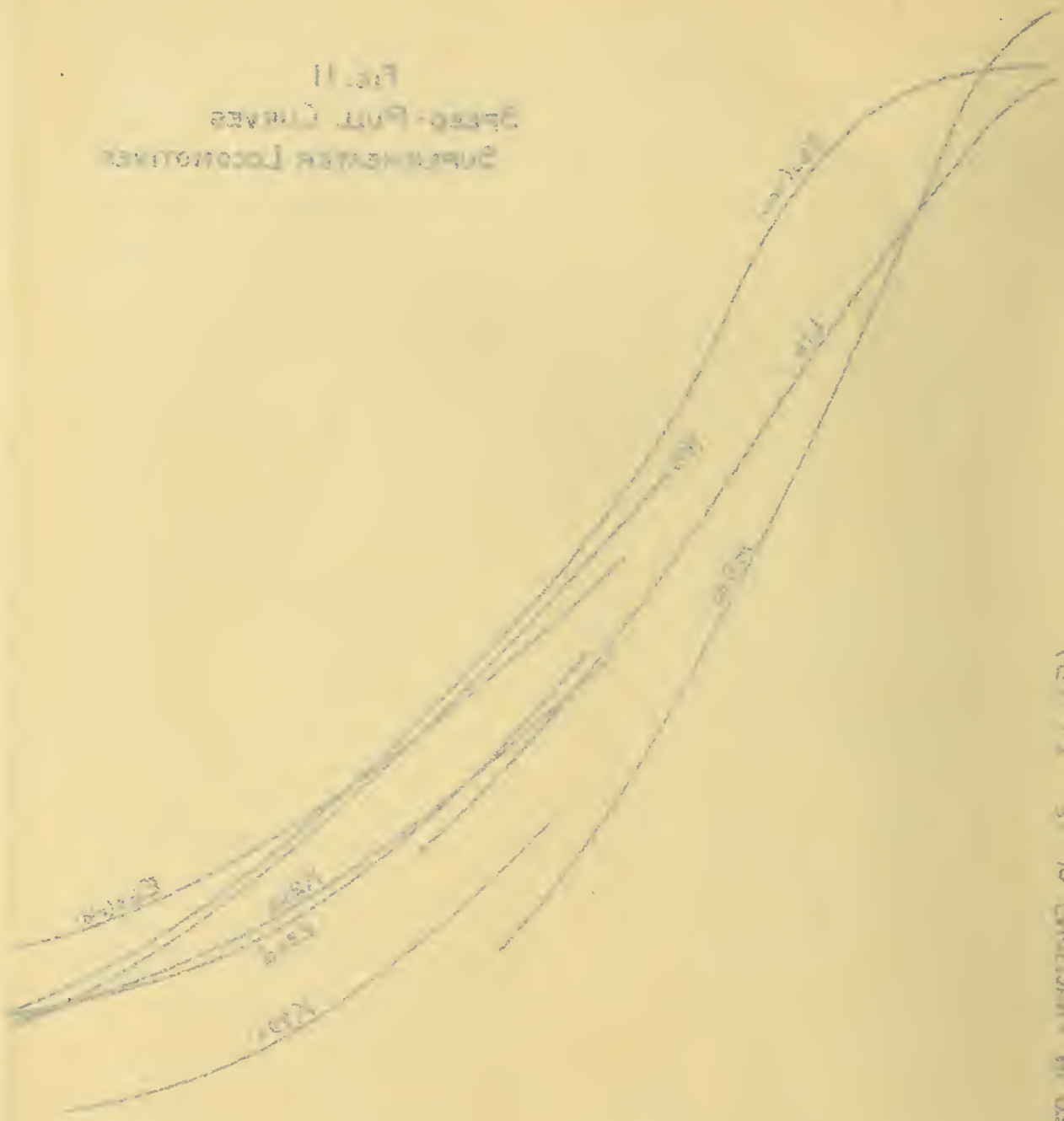


Fig. 11
Speed-Pull Curves
Superheater Locomotives
E6s (new)
L1s
K4s
H8sb
K2sa
E3sd
K29s
E6s (old)
.80
.70
.60
.50
.40
.30
.20
Cyld. T.E., expressed in fractions of To (pld²/D)
Piston Speed, F.P.M.
200
400
600
800
1000
1200



The incorrectness of the theories just stated. There is no ground for impeaching the theory, and no more data available, therefore a basis of comparison is sought which will be less affected by such irregularities as there may be.

The ordinates of Fig. 11 may be interpreted physically as the number of pounds of pull developed per pound of T_o or pld^3/D. As has been previously said, the tractive effort at any speed is governed by three main variables -- tractive effort rating, the steam available, and the efficiency of the steam distribution. A comparison on the basis of the tractive effort rating has proved unsatisfactory; the steam distribution variable in this portion of the problem has already been accounted for; only the evaporation remains. This variable may be given more prominence by replotting the curves, making the new ordinates "pounds of pull per pound of hourly evaporation." In the case of the testing-plant records, the evaporation for each test is known, but a comparison on this basis would provide no information applicable to road service. Therefore a particular rate of equivalent evaporation will be assumed as a base for each engine: not the absolute maximum attained in the laboratory tests, but a rate which represents the average evaporation of the three or four tests run at the highest power output. The adoption of such a standard for the boiler output provides a basis upon which road engines may be later compared.

In Fig. 12 the series of speed-pull curves is redrawn on this evaporation basis, and in this figure a satisfactory concordance is obtained. With the exception of one engine (the

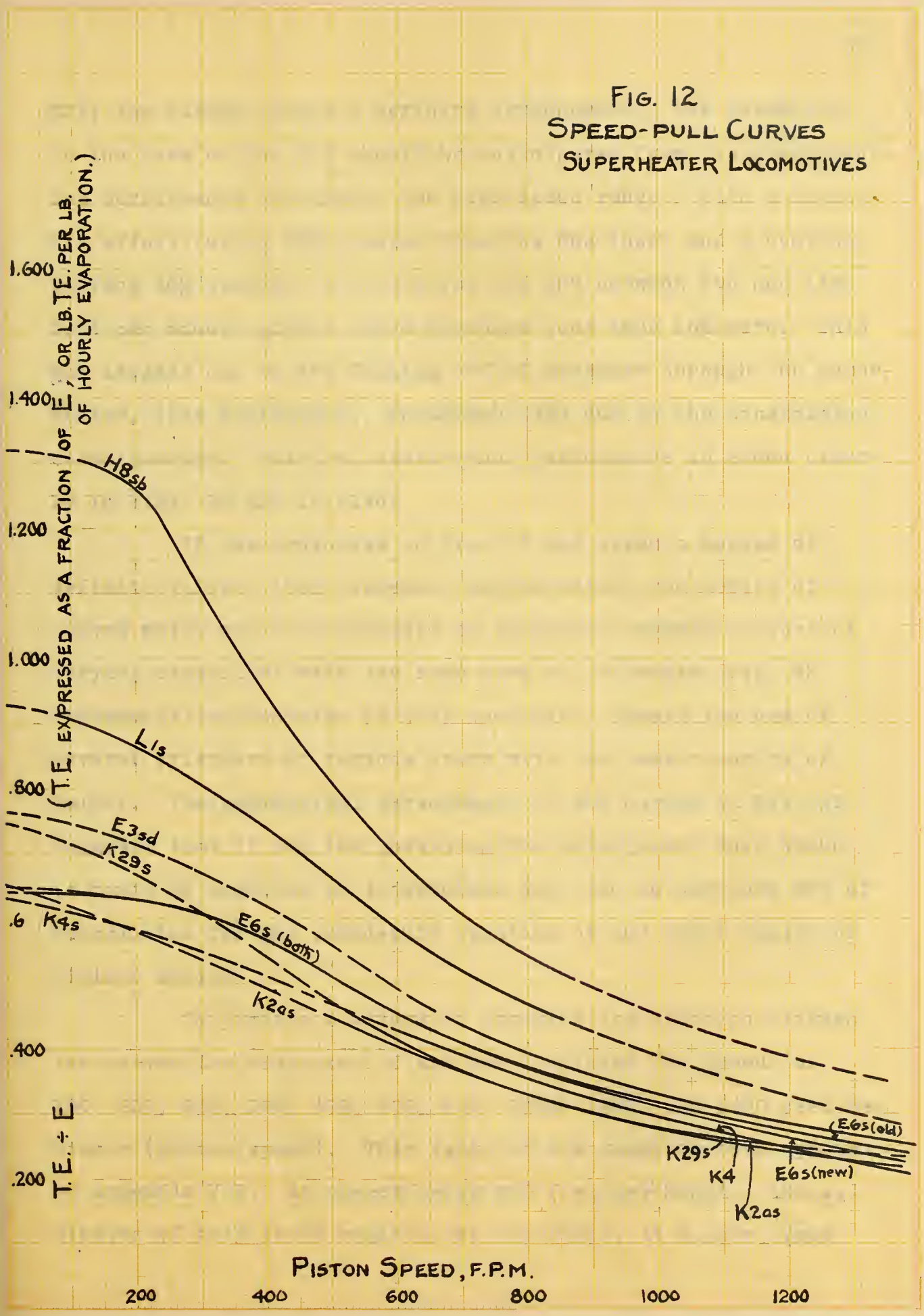
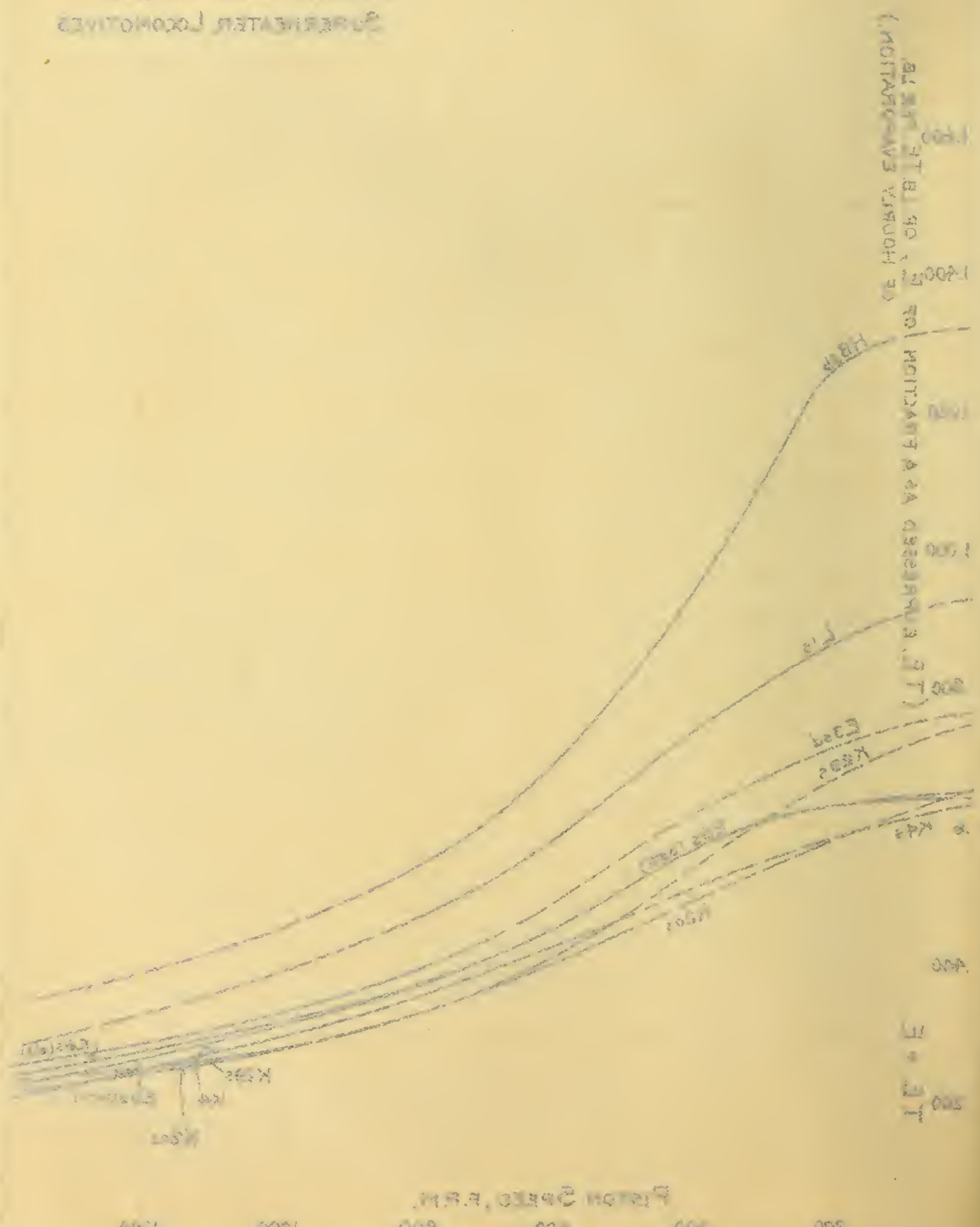


FIG. 12
SPEED-PULL CURVES
SUPERHEATER LOCOMOTIVES
T.E. ÷ E (T.E. EXPRESSED AS A FRACTION OF E, OR LB. T.E. PER LB. OF HOURLY EVAPORATION.)
1.600
1.400
1.200
1.000
.800
.6
.400
.200
H8sb
L1s
E3sd
K29s
K4s
E6s (both)
K2as
E6s (old)
K29s
K4
E6s (new)
K2as
PISTON SPEED, F.P.M.
200
400
600
800
1000
1200



K29) the curves follow a definite arrangement. The variation in the case of the K29 should be anticipated from its disappointing performance throughout the high-speed range. With a tractive effort rating 48% greater than the E6s (new) and a heating surface 60% greater, the pull for the K29 between 700 and 1400 feet per minute piston speed averages less than 10% more. This was largely due to the falling off of pressure through the superheater, (See Bulletin 21, Paragraph 149) due to the constricted steam passage. This relatively poor performance is shown clearly in Figs. 10 and 11 also.

If the ordinates in Fig. 11 are given a series of definite values, they represent approximately the series of curves which would be produced by equipping several boilers of varying capacities with the same size of cylinders; Fig. 12 represents the converse of this condition, namely the use of several cylinders of various sizes with the same capacity of boiler. The symmetrical arrangement of the curves in Fig. 12 suggests that if the law governing the arrangement were known, it would be possible to interpolate and find an accurate set of coordinates for the speed-pull relation of any other engine of similar design.

To provide a method of studying the relation between the curves the ordinates of all are tabulated for speeds of 100, 200, 250, 300, 400, 600, 800, 1000, 1200, and 1400 feet per minute (piston speed). This table in its complete form appears as Appendix V-D. At speeds below 600 feet per minute, the ordinates of only three engines are available; at higher speed

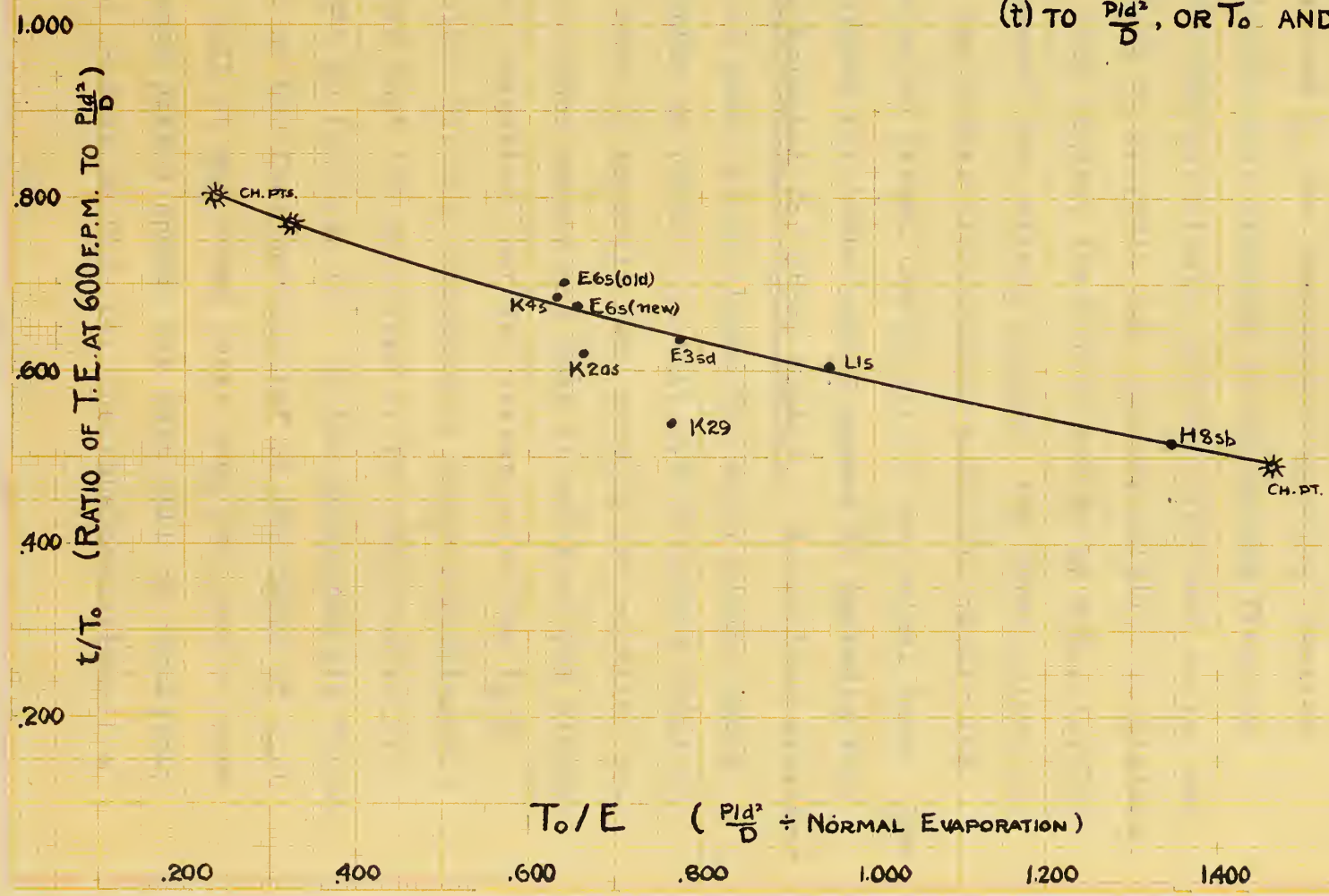
all of the curves may be used. Since at a speed of 600 feet all of the curves are actual experimental results (except the curve of the E3sd, which requires only a very slight extension), this speed serves as a convenient basing point for a further study of the relation of the other ordinates. In the second half of the table of Appendix V-D, the series of ordinates for each engine are calculated in percentages of the ordinate for that engine at 600 feet per minute. At speeds above 600 feet, the agreement among the percentages is very close for any given speed: the average percentage for each speed would vary only 3% (of the ordinate at 600 feet) from the highest and lowest percentages at 1400 feet per minute piston speed, and less at lower speeds. However, it will be noted that at all speeds except one, the E6s (old) has the highest percentage, and that all speeds except one the H8sb has the lowest, so a further investigation will be made, rather than adopt the average.

At speeds below 600 feet per minute, the percentage variation is much wider and an average is far from representative. Here the highest percentages are those of the H8sb and the lowest those of the E6s(new). The similarity of design of the two E6s engines (both heavy superheater Atlantics with unusually ample boilers) combined with the fact that the opposite extreme of the percentages is represented by a Consolidation type of average boiler capacity, suggests that there is a close connection between the ratio of cylinder power to boiler capacity and the variation of these percentages.

2. Relation Between Boiler Capacity and the Pull at 600 feet per minute. In order to fix the relation between the various curves of Fig. 12 two other relations may be determined. One is the position of the basing point, the ordinate for 600 feet per minute. (This will be referred to hereafter as "t"). The other relation is that between the other ordinates and "t". Of these two, the former is obviously dependent upon the relation existing between boiler capacity and cylinder size, as the higher curves are the freight engines with larger cylinders relative to their boilers, and the lower curves are those of the passenger engines with generous boiler capacity. This shows most clearly in the relative positions of the curves for the L1s and the K4s in Fig. 12. These engines have a boiler identical in design; the former having a value of T_o of 72100, and the latter of 52400 pounds.

In order definitely to evaluate the relation between boiler capacity and cylinder-power, the ratio representing this relation will be taken as the ratio of T_o to the equivalent evaporation assumed as a base for calculating the speed pull curves, or T_o/E. This is the value which, multiplied by a constant has already fixed the zero-speed point of the curves of Fig. 12. It will be noted that no assumptions are made concerning the capacity of the fireman or that the boiler was being driven to its largest output: the capacity used corresponds with that actually developed in the tests in question. One assumption must be made, however: that the steam generated was efficiently used; this was true of all of the tests except that of the K29,

FIG. 13
RELATION OF T.E. AT 600
(t) TO $\frac{PId^2}{D}$, OR T_o AND BOILER CAPACITY.



and the effects of this inefficiency show clearly throughout the investigation.

The information desired concerning the variation of the point "t" will be most conveniently given if we know how the variation of the ratio T_o/E affects the ratio between t, the pull at 600 feet, and T_o the theoretical starting tractive effort. The data for this relation is calculated from the values of the 600-feet-per-minute- ordinate for each engine and from the ratio T_o/E in Appendix V-E, and the relation is plotted in Fig. 13. In this figure, the points marked by small circles and labeled represent the various engines; the three points at the extremes of the curves are checking points to verify the extension of the curve beyond the limits of the actual data. The data and calculations for these points appear in Appendix V-F.

3. <u>The Relation for Other Speeds.</u> A relation similar to that of Fig. 13 could be developed for every piston speed, but it seems simpler to produce another set of curves based on Fig. 13 and the data in Appendix V-D, showing the relation between the pull at other speeds and that at 600 feet per minute. It was previously suggested that the variation of the T_o/E ratio is responsible for the variation of these percentages, and this indicates that the abscissae of the comparative plot should be the same as for Fig. 13. If the ordinates are "fraction of the pull at 600 feet per minute" or the ratio of t_s/t where t_s is the pull at any other speed than 600, and a curve is plotted for each piston speed, the data will be completely presented. This is done in Fig. 14, the interpretation of a

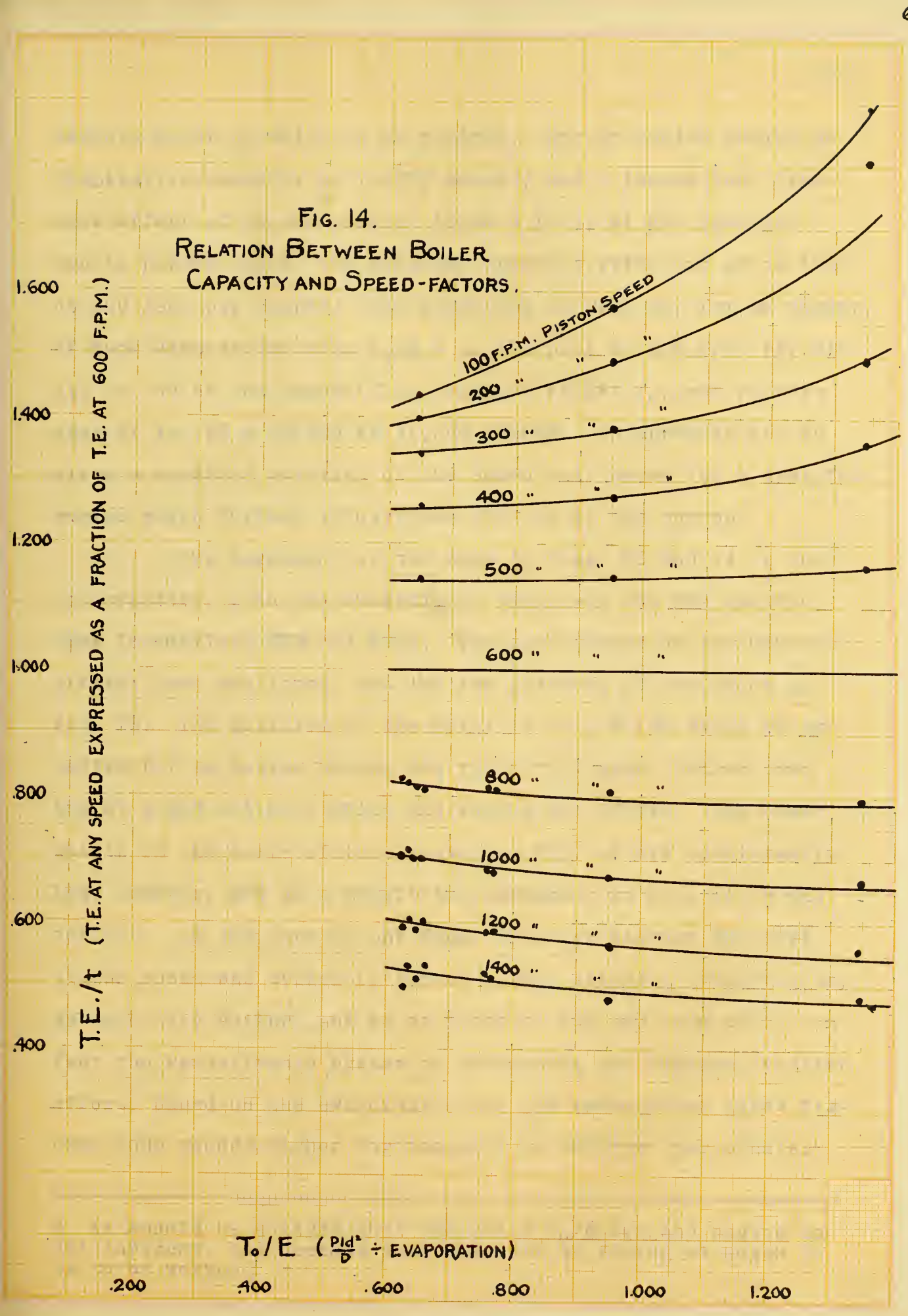


FIG. 14.
RELATION BETWEEN BOILER CAPACITY AND SPEED-FACTORS.
100 F.P.M. PISTON SPEED
200 "
300 "
400 "
500 "
600 "
800 "
1000 "
1200 "
1400 "
1.600
1.400
1.200
1.000
.800
.600
.400
T.E./t (T.E. AT ANY SPEED EXPRESSED AS A FRACTION OF T.E. AT 600 F.P.M.)
T₀/E ($\frac{Pld^2}{D}$ ÷ EVAPORATION)
.200
.400
.600
.800
1.000
1.200

certain point of which is as follows. For an engine having an evaporative capacity of 70,000 pounds, and a theoretical tractive effort of 70,000 pounds, ($T_o/E = 1.0$), at 800 feet per minute piston speed, the cylinder tractive effort is .8 of that at 600 feet per minute. Now referring to Fig. 13, for an engine of such dimensions* that $T_o/E = 1.$ the pull at 600 feet per minute is .59 of the theoretical tractive effort T_o, and in this case it is .59 x 70,000 or 41,300 pounds. In Appendix V-G is given a complete solution of the speed pull curve for a specific engine which further illustrates the use of the curves.

The agreement of the data in Figs. 13 and 14 is very satisfactory, with two exceptions; these are the two Pacific type locomotives K29s and K2sa. The performance of the K29s has already been mentioned, and the low position of its point in Fig. 13. The position of the point is that which would be expected for an engine having 24½ x 28" cylinders, rather than the 27 x 28" set with which the engine was fitted. The other points on the curve of this engine in Fig. 13 are consistently low, however, and as a result the agreement in Fig. 14 is excellent. In the case of the K2sa, the test nearest 600 feet piston speed was evidently run at a rate slightly under the capacity of the boiler, and so of those at 800 and 1000 feet. In fact the Pennsylvania system of estimating the maximum tractive effort, based on the evaporation and the water-rate, gives figures 2000 pounds higher for the pull at 600 feet per minute:

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* It should be noticed that the ratio T_o/E for any engine is not inherent, but depends upon the rate of firing at which it is to be worked.

this figure would place the K2as point exactly on the curve in Fig. 13. As a means of justifying the steps taken in generalizing this data, Appendix V-H presents comparisons of the actual and calculated curves for two of these engines, showing the best and poorest agreement between the actual and the calculated curves.

VI. APPLICATION OF THE PROCESS TO LOCOMOTIVES USING SATURATED STEAM.

The analysis developed in the preceding section is equally applicable to locomotives using saturated steam, provided the data are available. Testing plant records are available for twelve locomotives using saturated steam in simple cylinders. Of these, five fall into the class referred to as "A", having piston valves, outside valve-gears, outside steam pipes, and all other late improvements except the superheater; one is of class B, having a piston valve but no other improvements; the remain six are of class C, having slide valves. A list of these engines with their principal dimensions will be found in Appendix VI-A.

The data in most of the cases are fragmentary, and for the majority of the engines they cannot be used in the original form. In Appendix VI-B will be found a presentation of the original data, with the methods used in rendering the figures available for the purpose of the present writing. Based upon the material thus developed, the speed-pull curve for each engine has been plotted in the same terms as previously employed: that is, pounds pull per pound of hourly evaporation, and piston speed. The curves for the engines of Classes A and B appear in Fig. 15: for those of Class C in Fig. 16, the data for these

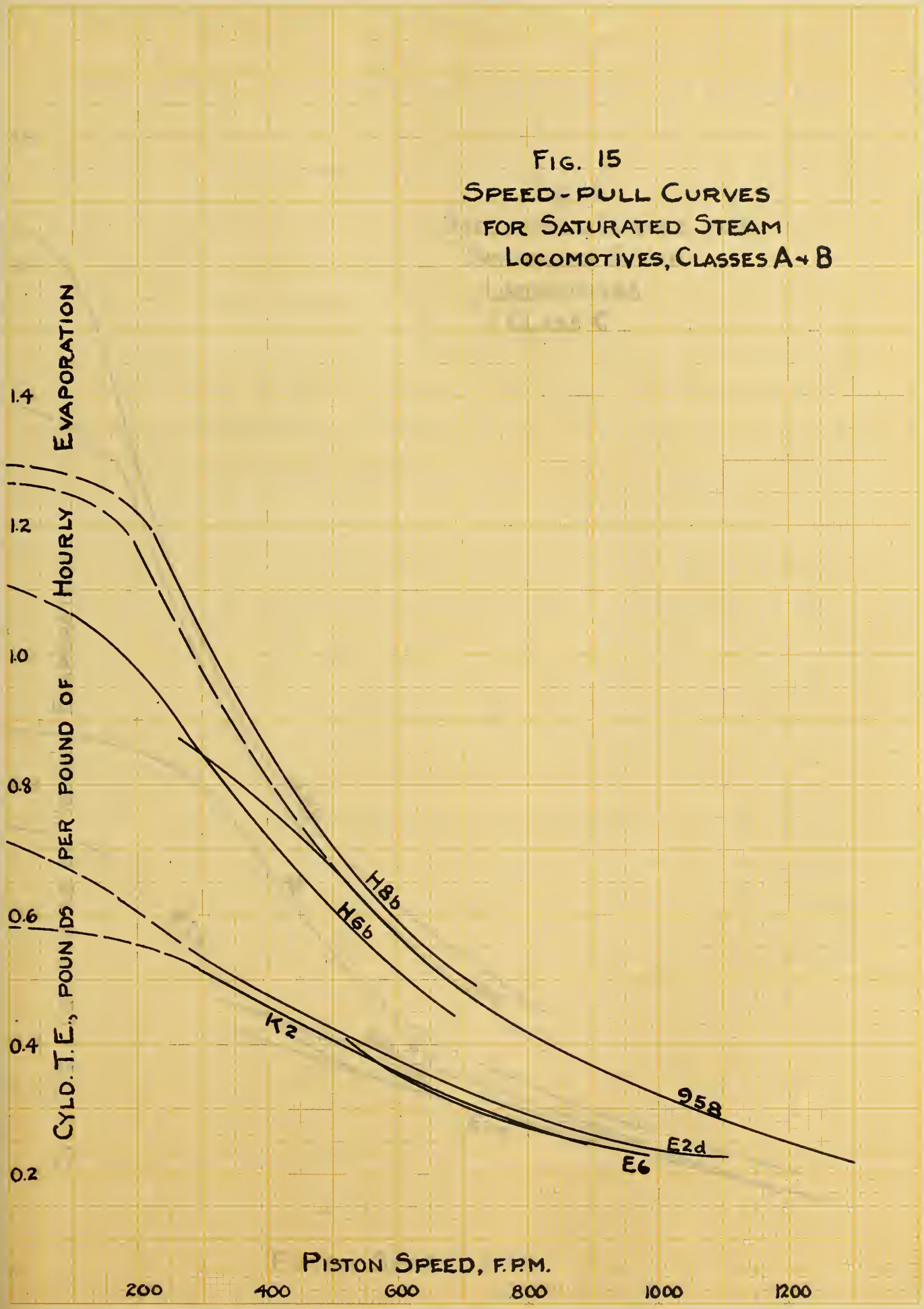
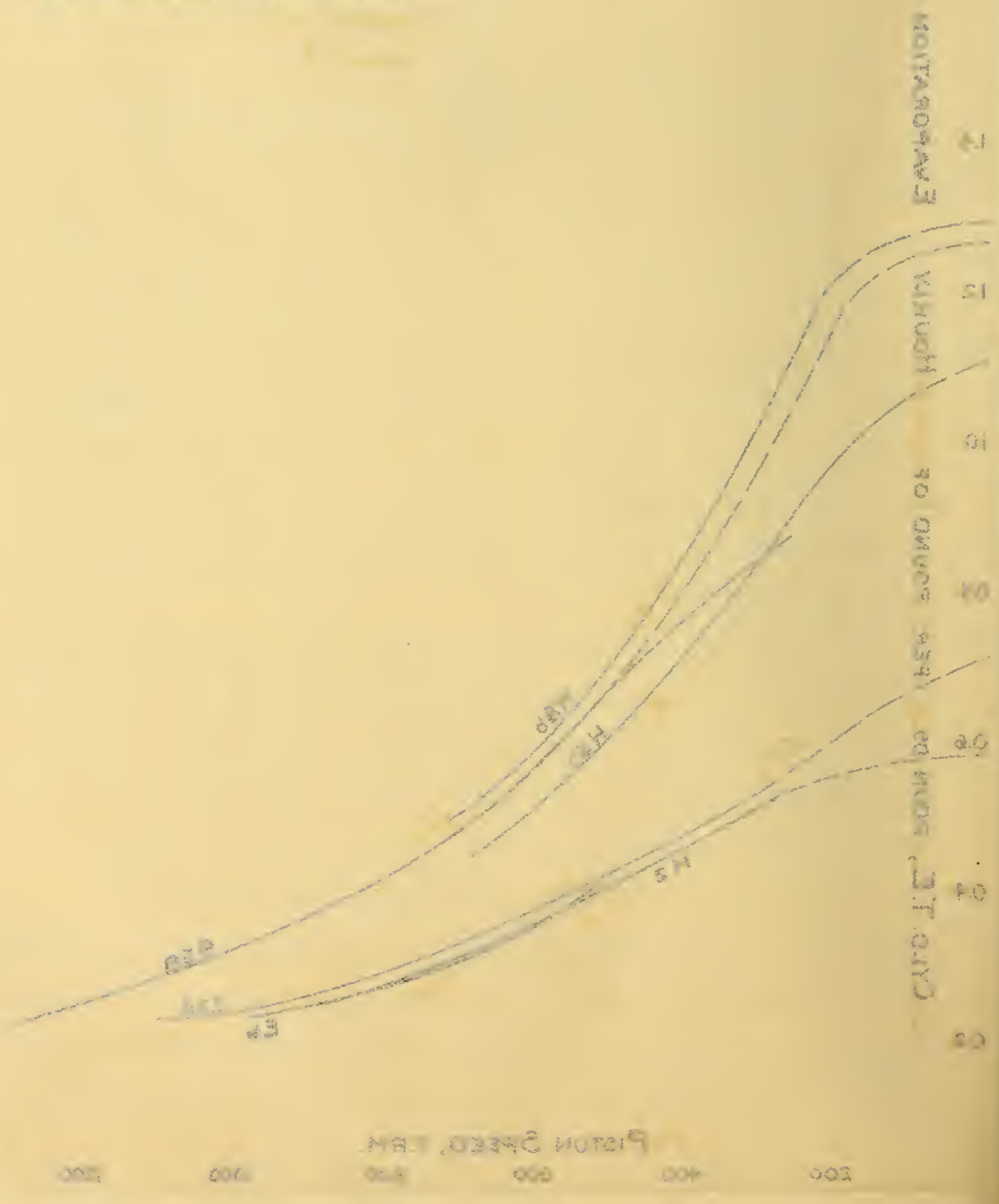


Fig. 15
Speed-Pull Curves
for Saturated Steam
Locomotives, Classes A + B
Cyld. T.E., Pounds per Pound of Hourly Evaporation
1.4
1.2
1.0
0.8
0.6
0.4
0.2
H8b
H6b
K2
958
E2d
E6
Piston Speed, F.P.M.
200
400
600
800
1000
1200


Piston Speed, FPM
200
400
600
800
1000
1200
Cut-off, Fraction of Stroke
0.0
0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8
0.9
1.0
Expansion Ratio
1.4
1.5
1.6
1.7
1.8
1.9
2.0
2.1
2.2
2.3
2.4
2.5
2.6
2.7
2.8
2.9
3.0
3.1
3.2
3.3
3.4
3.5
3.6
3.7
3.8
3.9
4.0
4.1
4.2
4.3
4.4
4.5
4.6
4.7
4.8
4.9
5.0
5.1
5.2
5.3

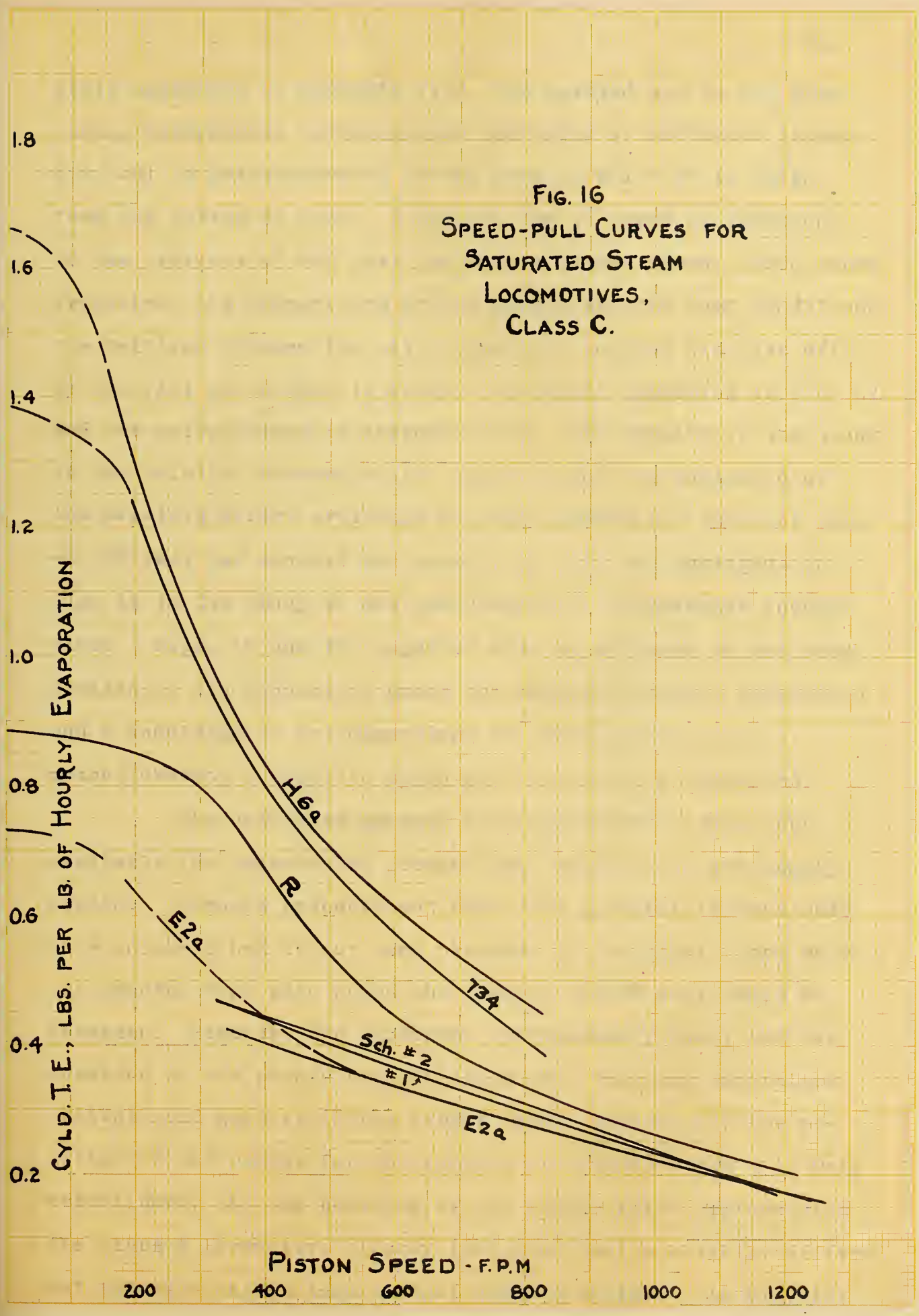
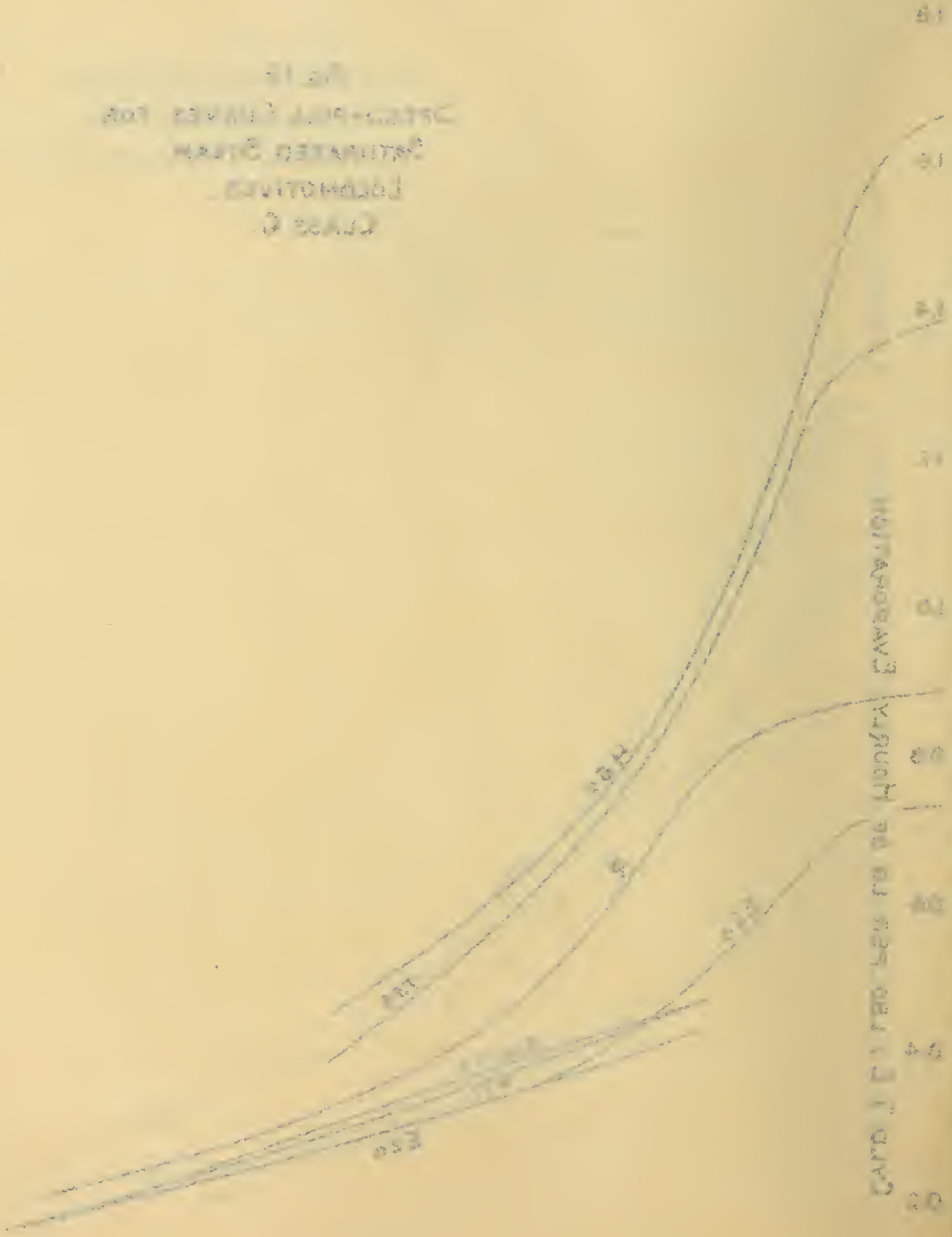


Fig. 16
Speed-pull Curves for
Saturated Steam
Locomotives,
Class C.
1.8
1.6
1.4
1.2
1.0
0.8
0.6
0.4
0.2
Cyld. I. E., lbs. per lb. of Hourly Evaporation
H6a
R
E2a
134
Sch. #2
#1
E2a
200
400
600
800
1000
1200
Piston Speed - F.P.M



plots appearing in Appendix VI-C. As carried out in the preceding superheater investigation the table of ordinates (Appendix V-D) is presented both in the form in which it is taken from the curves of Figs. 15 and 16, and in terms of fractions of the ordinate at 600 feet per minute piston speed. From these ordinates, the proportions of the engine and the test conditions, the relation between the boiler capacity and the tractive effort at 600 feet per minute is found, the result appearing in Fig. 17, and the calculations in Appendix VI-D. The results of the study of the relation between boiler capacity and the variation of the tractive effort available at other speeds (in terms of that at 600 feet per minute) are plotted in Fig. 18, analogous to Fig. 14 in the study of the performance of superheater locomotives. Figs. 17 and 18, together with an estimate of the evaporation of the locomotive under the proposed service conditions and a knowledge of the dimensions of the engine furnish a method whereby a complete speed-pull curve may be produced.

The data used are much less satisfactory than that available for superheater locomotives, as has been previously stated. It would probably not have been possible to have made this analysis had it not been preceded by the other, from which the general form into which the results should fall could be foreseen. However, the agreement is reasonably good, and the location of the points shows clearly that the same general relations hold good as in the former case. In Fig. 17, the position of the curves for the engines of classes A and C is well established, and the position of the single point representing the class B locomotive clearly indicates that a slightly different relation exists than that of Class A engines. In Fig. 18,

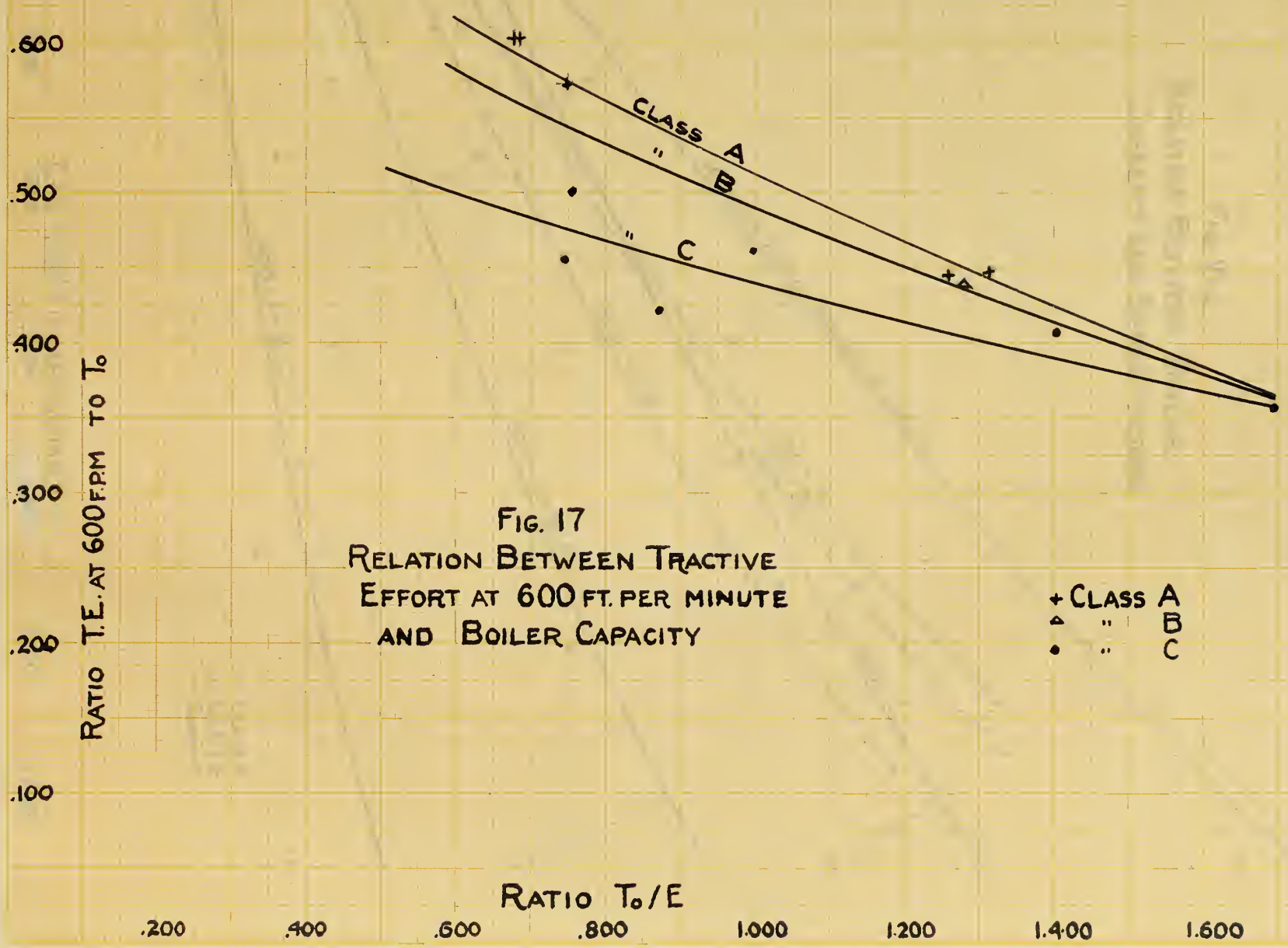


FIG. 17
RELATION BETWEEN TRACTIVE EFFORT AT 600 FT. PER MINUTE AND BOILER CAPACITY

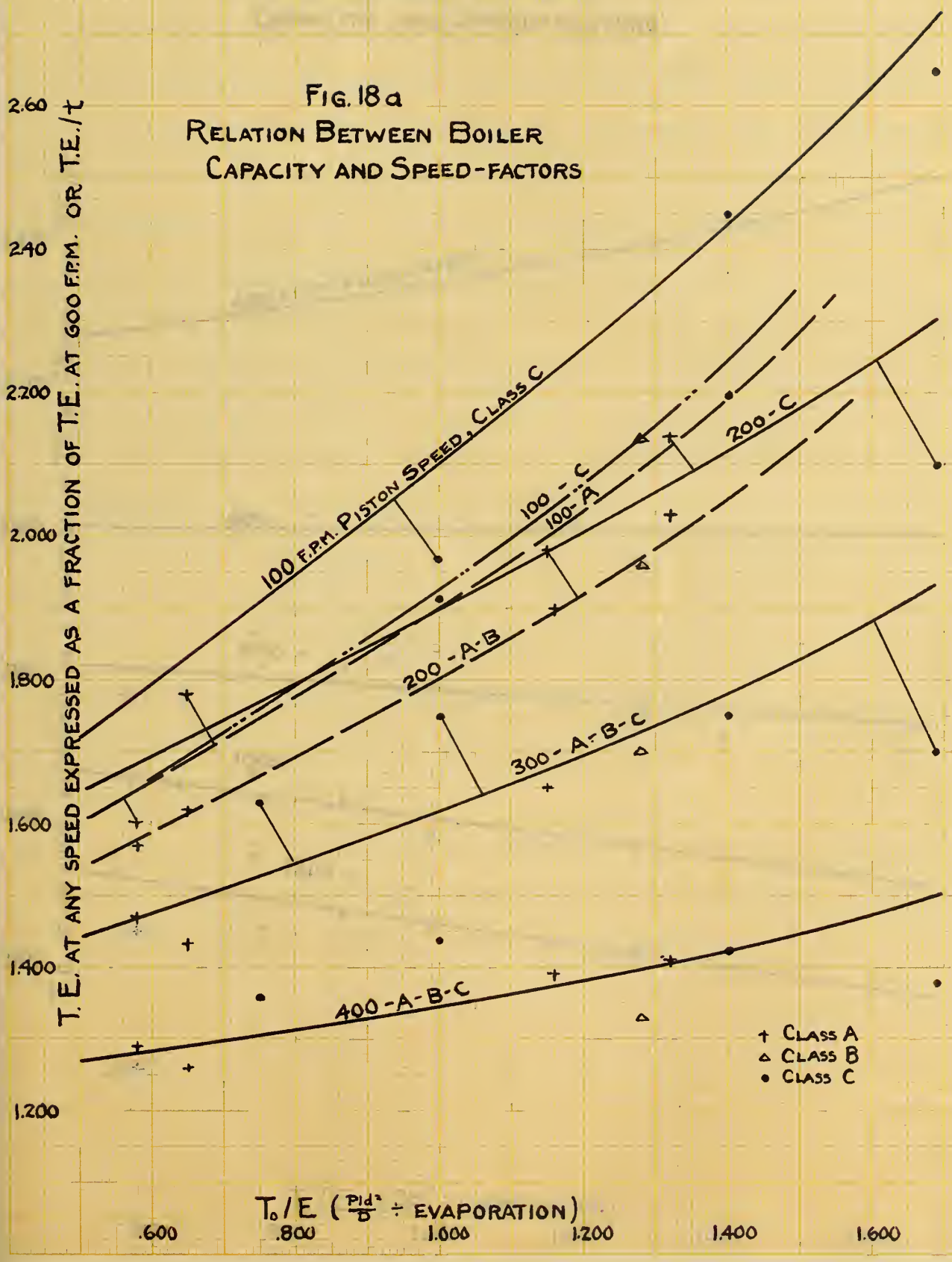
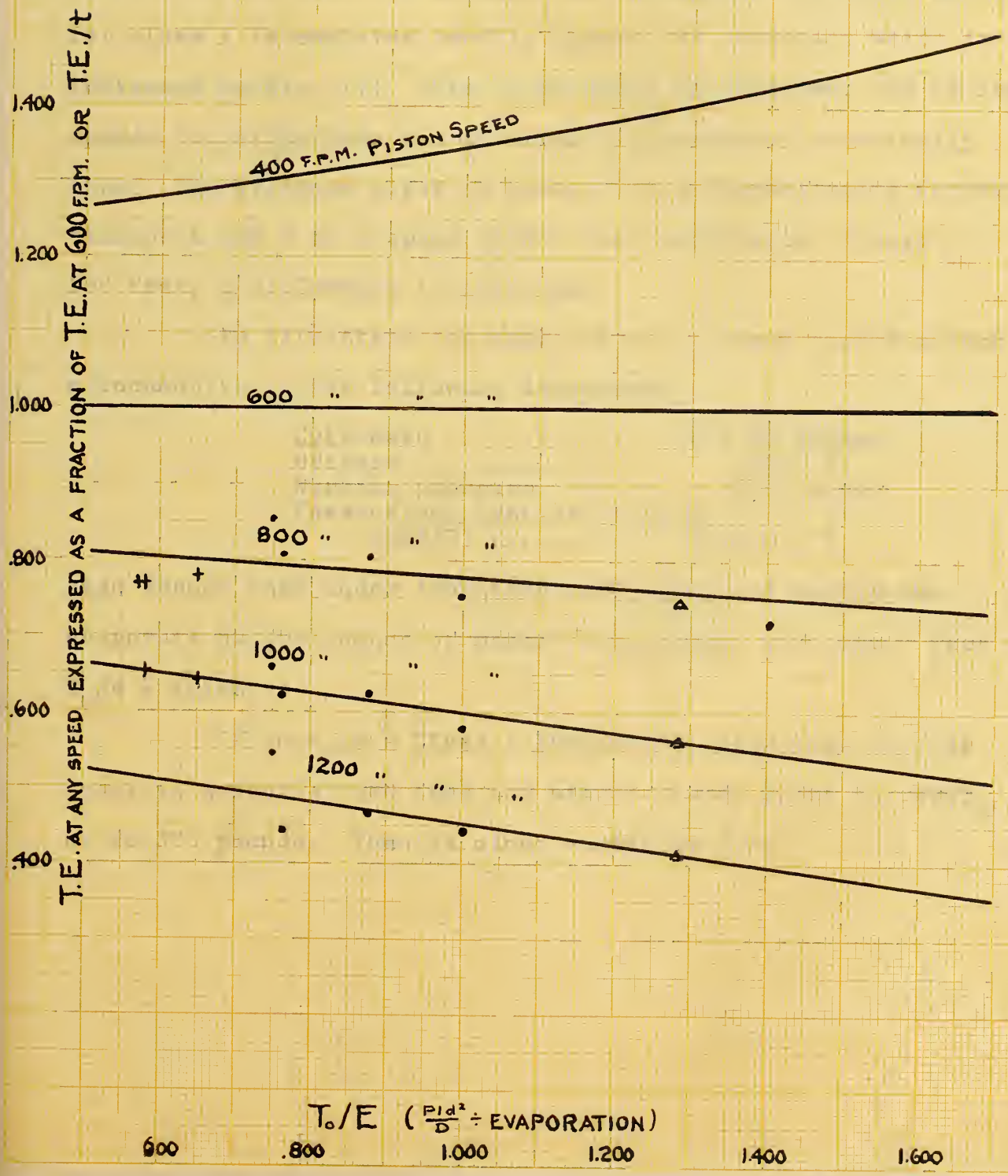


FIG. 18a
RELATION BETWEEN BOILER CAPACITY AND SPEED-FACTORS
T.E. AT ANY SPEED EXPRESSED AS A FRACTION OF T.E. AT 600 F.P.M. OR T.E./t
2.60
2.40
2.200
2.000
1.800
1.600
1.400
1.200
100 F.P.M. PISTON SPEED, CLASS C
100 - C
100 - A
200 - C
200 - A - B
300 - A - B - C
400 - A - B - C
+ CLASS A
△ CLASS B
• CLASS C
T₀/E (Pld²/D ÷ EVAPORATION)
.600
.800
1.000
1.200
1.400
1.600

Fig. 18b
Relation Between Boiler Capacity and Speed-factors



at speeds higher than 200 feet per minute there is nothing in the evidence to warrant differentiation of the classes: at these speeds therefore the variation in tractive effort due to the steam distribution characteristics is the same proportionally as that at 600 feet per minute. At low speeds the lower curves for class A locomotives tend to reduce the variation which is indicated by Fig. 17: this is as would be expected, for at low speeds the difference due to steam distribution is obviously less. The evidence gives no ground for differentiating between classes A and B at a speed of 200 feet per minute, though at 100 feet, a difference is indicated.

To illustrate the application of these curves assume a locomotive of the following dimensions:

Cylinders	24 x 30 inches
Drivers	60 "
Working pressure	200 pounds
Theoretical tractive effort (pld/D)	57,600 "

Also assume that under the given conditions the boiler can evaporate 52,500 pounds of water (equivalent) per hour. Then $T_o/E = 1.100$.

If this is a Class A locomotive, from Fig. 17, the tractive effort at 600 feet per minute piston speed is $.490T_o$, or 28,300 pounds. Then at other speeds we find:

Piston Speed (ft. per minute)	Ordinate	Tractive Effort (Pounds)	(Miles per hour)
100	1.950	54800	3.56
200	1.860	52500	7.12
300	1.650	46500	10.18
400	1.380	39400	14.24
600	1.000	28200	21.36
800	.775	21900	28.48
1000	.590	16800	35.60

As a Class B engine: the tractive effort at 600 feet is .470 T_o or 27,000 pounds, and at other speeds we have:

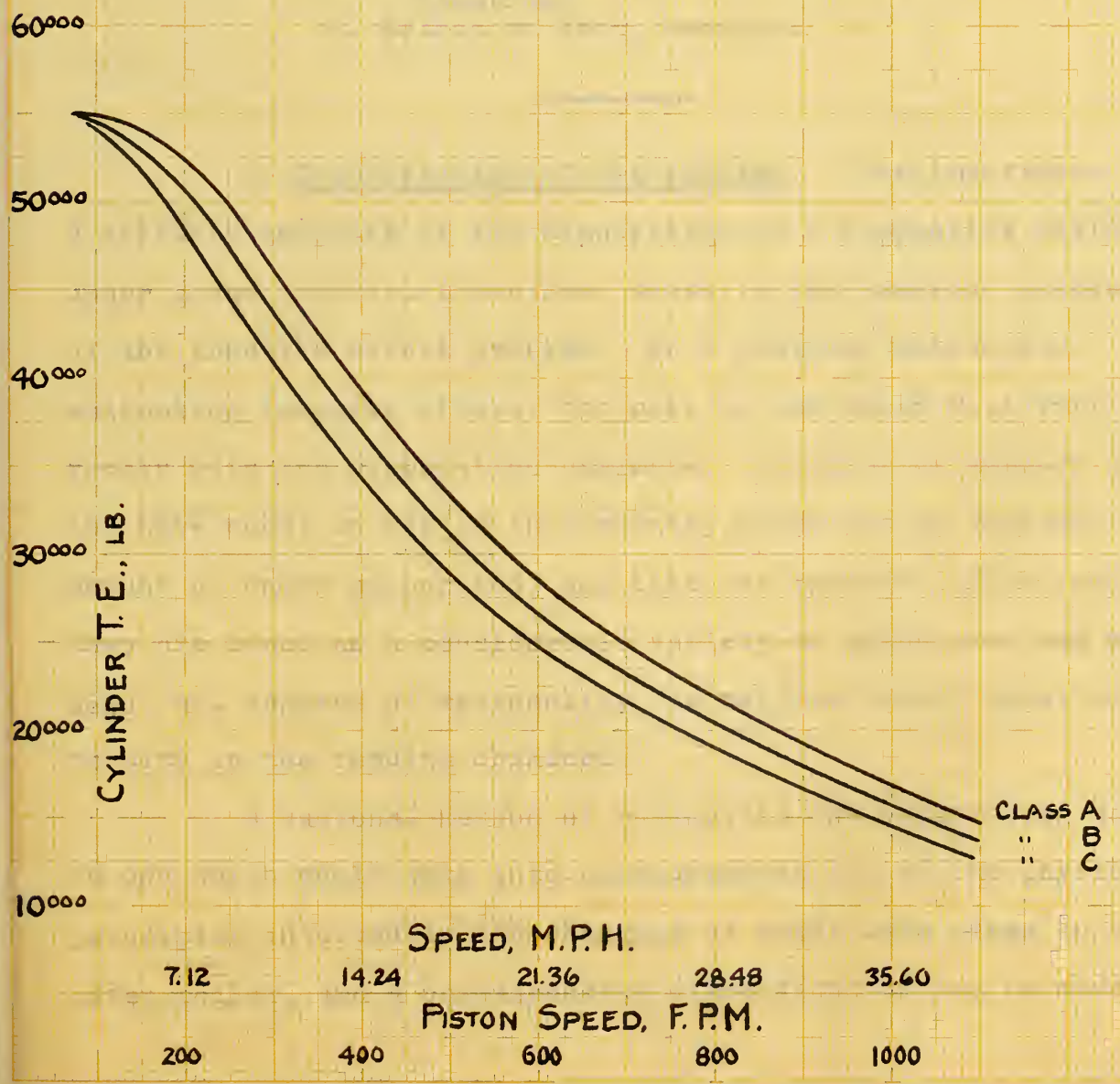
100	2.000	54000
200	1.840	49800
300	1.650	44500
400	1.380	37500
600	1.000	27000
800	.775	21000
1000	.590	15900

As a Class C engine: the tractive effort at 600 feet is .425 T_o or 24,500 pounds. At other speeds we have:

100	2.200	54000
200	1.960	48000
300	1.650	40500
400	1.380	33800
600	1.000	24500
800	.775	19900
1000	.590	14500

These curves are presented in Fig. 19.

Fig. 19
Effect of Steam Distribution on Tractive Effort.



VII. METHODS OF ESTIMATING BOILER PERFORMANCE.

1. The Variables of the Problem.
2. Development of the "Coal Factor".
3. Development of the "Equivalent Grate".
4. Comparison of Test Data on the Equivalent Grate Basis.
5. Discussion of Preliminary Investigations.
6. Method of Using Results.

1. The Variables of the Problem. The importance of a reliable estimate of the evaporation of a locomotive boiler under given conditions has been shown in the previous studies of the tractive effort problem. In a rational method for estimating tractive effort, the pull at any speed will vary directly with the evaporation estimate. A number of methods are (or have been) in use in this country which aim to forecast the amount of water evaporated, and like the tractive effort methods they are based on a considerable variety of principles and possess all degrees of rationality, as well as almost equal diversity in the results obtained.

A rational method of estimating the evaporation would be one which would take into consideration all of the physical principles involved in the changing of water into steam in a given boiler, and a corresponding classification may be made to

cover semi-rational and empirical methods. It may be said, however, that there has been no absolutely rational method produced, and the use of a strictly empirical method (which in this case would involve the assumption of a constant evaporation rate for all conditions, by analogy to the tractive effort method classification) gives such obviously crude results that it need not be further considered. The semi-rational methods which remain may be defined as those which take into consideration a few or most of the physical variables, as shown by their effects in actual test cases. The value of any such method depends on three things:

The number and importance of the variables considered.
The correlation and evaluation of these variables.
The agreement of the estimate with actual results.

There are two classes of variables which enter into a consideration of boiler capacity -- those relating to the design of the boiler, and those relating to the fuel used. Each of these is subdivided as follows:

Fuel Variables:
- Quantity of fuel fired
- Quality of fuel

Design Variables:
- Total water heating surface
- Length, diameter and spacing of tubes
- Superheating surface
- Firebox heating surface, including arch tubes
- Grate area.

A third variable might be added, namely, the water used. The only effect of water on the efficiency is that due its scale forming properties, and the effect of scale has already been

accurately evaluated by other investigators. The final performance estimate may be modified for the presence of scale if desired, but this need not be regarded as a part of the process itself.

The variables mentioned are of widely differing degrees of importance, the total evaporation of any boiler varying more nearly directly with the amount of coal fired than with any of the others. The effect of tube diameter and spacing is the least important. The familiar boiler performance curves show that the evaporation increases with the quantity of coal fired throughout the working range of the boiler, but that each increment of coal is less efficient. A superficial consideration of the effect of varying the quality of coal, the other conditions remaining the same, would suggest that the variation in evaporation rate would correspond directly to the variation in the B.t.u. value of the coal, and this assumption has been made by several investigators. This question will be discussed at length later.

Next in importance to the total fuel fired is the total heating surface. In a general way the total heating surface is an index of the capacity of the locomotive, a large heating surface anticipating a high rate of firing. With the same total coal fired, an increase in the heating surface will also give increased capacity since the efficiency of the boiler is higher at lower rates of coal consumption per square foot of heating surface. The design of the tubes has some effect on the evaporation, the length being the most important element.

According to Mr. Cole's investigation,* with other conditions equal, the evaporation per square foot of heating surface of the shortest tubes in general use, is about thirty per cent. greater than that of the longest. An evaporation per square foot of heating surface five per cent. greater may be expected from 2-1/4-inch tubes than from 2-inch tubes, and the tubes with the widest spacing should give about fifteen percent. more evaporation than those with the least spacing.** Since the evaporative capacity of the direct heating surface (fire-box surface and arch tubes) is much greater -- from five to eight times -- than that of the tube surface, it follows that the ratio of the firebox heating surface to the total surface -- or rather the ratio of the grate area to the total heating surface, since the firebox surface in ordinary design is a direct function of the grate area -- has a large effect on the evaporation rate of the boiler as a whole. A large grate will produce a high total evaporation, a high rate or evaporation per square foot of heating surface, and a low rate per pound of coal, for it is a thoroughly established law of boiler performance that capacity and efficiency are opposing elements -- one is obtained at the expense of the other.

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* This figure and the two following percentages are taken from comparison of the figures in Mr. Cole's bulletin "Locomotive Ratios" (No. 1017, published by the American Locomotive Co.) The investigation is discussed later.

** Note that all of these percentages apply to the tubes alone. The tube evaporation forming about two-thirds of the total for the boiler, the effect on the boiler as a whole may be evaluated by reducing each percentage one third.

Several methods which have been used for estimating boiler capacity in connection with tractive effort estimates were mentioned in Section II, but for convenience their characteristics may be here reviewed.

The Goss and Kiesel methods assume constant evaporation rates and direct variation of the total evaporation with the heating surface. The implication of the Baldwin method is the same.

The A.R.E.A. method takes into account the quantity and quality of the coal, and the total heating surface. The results are very conservative.

The Williamson method implies either a constant boiler efficiency if the rate of firing is taken at one pound of coal per square foot of heating surface per hour, or the ignoring of the rate of firing though taking into account the quality of the coal.

The Houston method assumes a maximum rate of firing of 1-3/4 pounds of coal per square foot of heating surface per hour, and assumes corresponding evaporation rates for various grades of coal. If the total coal fired amounts to over 5000 pounds per hour, the amount is reduced to 5000, and the evaporation is reduced in direct ratio to the reduction of the amount of coal.

There are three other methods of making the evaporation estimate well worthy of attention, which were not previously considered. The most important of these is that of the bulletin "Locomotive Ratios" published by the American Locomotive Co.

In this bulletin Mr. Cole has made a thoroughly rational investigation of the evaporative efficiency of the heating surface, considering every variable named. However, in this discussion, which is based upon the results from tests in which a practically constant quality of coal was used, and that the best in use for locomotive service, the author fails to take into account in any way the variation of firing rates or quality of fuel. Therefore the use of this method involves some assumption as to the working of the engine, whether "to the limit", "all-day performance" or some intermediate rate. Use of the method on the part of the writer has shown that for small engines -- say for those having up to 2500 square feet of heating surface, the estimate represents a "close-to-the-limit" performance; for large engines the rate is rather moderate. Mr. Cole's figures are based on the use of coal of 14500 to 15000 B.t.u., and it has also appeared that by simply reducing the estimate in the direct ratio of the quality of the coal, the method gives results well in harmony with actual performance under good average conditions for engines of moderate size.

The first method proposed by Mr. Shurtleff in connection with the tractive effort estimate of the A. R. E. A. provided a chart showing the relation between ratio of heating surface to grate area, firing rate, and evaporation per pound of coal. By correcting the estimate given for the quality of the coal as is done in the adopted A. R. E. A. method, all of

the major variables are accounted for.*

In "Locomotive Operation", p. 471, Mr. Henderson gives a chart for estimating evaporation based on the work of Mahler (see Kent's Pocket-book, 1906 ed., p. 634). This estimate does not consider the design of the boiler in any way, but takes account of the variation in the quality of the fuel.

For reference as to the results of the estimates, in Appendix VII-A, the evaporation of a certain Pennsylvania engine has been calculated by each of the methods named. The results are as follows:

Summary of Evaporation Estimates.

Method of Estimating	Evaporation Actual.	per Hour Equiv't.	Firing Rate lb. per hr.
Tests	42,600 lb.	35,500 lb.	5530 lb.
A.R.E.A.	31,600 lb.	26,300 lb.	5530 lb.
Henderson	56,200 lb.	47,000 lb.	5530 lb.
Shurtleff	36,000 lb.	30,000 lb.	5530 lb.
Cole	38,600 lb.	32,200 lb.	Indefinite
Tests	37,500 lb.	31,250 lb.	4300 lb.
Houston	30,800 lb.	25,700 lb.	4300 lb.
Tests	25,200 lb.	21,000 lb.	2450 lb.
Williamson	29,000 lb.	24,000 lb.	2450 lb.

An evaporation factor of 1.2 was assumed. The coal used was of 14,500 B.t.u. The evaporation during tests under the same conditions for which the estimates are valid was 42,600 pounds equivalent, or 35,500 pounds actual, per hour for 5530 lb. coal.

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* See Proc. A.R.E.A., Vol. 12, pt. 1, p. 631, 709. Also given by Williamson, Railway Age Gazette, Vol. 52, (1912) p. 686.

2. Quality of the Fuel. With the foregoing survey of the conditions of the problem, we may proceed with the study. The first variable which may be conveniently disposed of is that of the quality of the fuel. Practically all of the locomotive test data available are based on fuel of very uniform quality -- Pennsylvania bituminous coal. The only exceptions are the tests on the Purdue and Illinois plants. Pennsylvania coal was used in the St. Louis tests, also in some of those on the Purdue plant. However, the evaluation of this variable is desired, rather than its elimination. What is the effect of the thermal value of coal on evaporation, other conditions being equal? Kent, Barrus and other writers on boiler economy give data from which conclusions may be drawn; the work of Mahler previously referred to is also available, and Dr. Goss gives in "Locomotive Performance" curves showing the evaporative values of various coals. All of this data must be used in combination with so many assumptions that little weight can be given to the results. There are two satisfactory sources however: the reports of the United States Geological Survey Tests at St. Louis*, and a paper by Mr. A. Bement before the Western Railway Club, April 20, 1909. The former reports cover tests of seventy-eight different coals in two almost identical boilers, the firing rates being very uniform. The latter reports the result of a series of trials on a stationary boiler in which the coal fired was mixed with increasing percentages of ash until the evaporation became nil.

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* Technical Paper No. 48, especially Part 2.

The data from these two sets of experiments and also that of Mahler are reduced to a comparable basis as follows:

For the St. Louis tests of the United States Geological Survey, the B.t.u. of the coal and evaporation per pound of coal are given in the report on the basis of dry coal. In order to make the range of B.t.u. value wider, the theoretical value of the coal has been calculated to a basis of coal as fired, and an average curve is drawn for all of the tests to show the relation between the thermal value and the evaporation rate.

For the Bement Tests, from the figure given in the report, the values are altered to terms of B.t.u. of coal as fired and evaporation per pound, a logical assumption having been made for the fuel value of the coal knowing the locality whence it came, and for the B.t.u. remaining in the ash.

The same results are produced from the Mahler tests by completing the analysis, and calculating the results given to the proper basis. All data and calculations appear in Appendix VII-B.

In Fig. A-7 the resulting curves are plotted. It is of no moment whatever that the curves vary in form and position, since they relate to different types of boilers. The following table is suggestive of the information desired:

Experimenter and Type of Boiler Used.	Equiv. Evaporation per pound of coal Using coal of thermal value of 10000 B.t.u.	12000 B.t.u.	Increase
Mahler (?)	10.00	12.5	25.0%
U.S.G.S. (Heine)	6.10	7.62	24.9%
Bement (B. & W.)	6.00	7.45	24.2%

It will be noticed that under the usual assumption that the evaporation increases directly with the heating value, the increase should have been 20% in place of almost uniformly 25%. Coal of a value of less than about 9000 B.t.u. (as fired) is of no interest in locomotive service, and the approximation of the valuable portion of the curve to a straight line suggests that a cumbersome method of correction may be avoided by the use of a straight line representing the curve through the useful range. These lines are drawn in Fig. A-7, and it will be noticed that both lines have 3000 B.t.u. as their intercept. If this value be accepted, the correction factor for estimating the evaporation for another grade of coal when that for a certain grade is known is

$$F = (B' - 3000) \div (B - 3000)$$

where B is the thermal value of the coal for which the rate of evaporation is known, and B' the B.t.u. of the coal for which the rate is desired. In the absence of conflicting evidence this correction will be used in the further study of the evaporation problem. The assumption thus made is equivalent to the conclusion that a certain amount of the B.t.u. in the coal is unavailable for evaporative purposes, and that the evaporation will vary directly with the number of heat units in excess of this unavailable quantity.

3. <u>An Index of Boiler Capacity</u>. Returning to the list of variables on p. 80, there remain to be considered the quantity of fuel to be used, and all of the variables grouped under design features. It is possible to take into account two

variables or factors conveniently in a chart, in addition to the rate of evaporation. Eight remain in the problem however, therefore further justifiable methods of combination and elimination of six of these will be sought. First, the length of tubes is an index of their diameter, and the spacing is a matter of very small importance within the range of design generally adopted, so that diameter and spacing may be ignored. Second, if the grate area is taken into account, the direct heating surface which utilizes the heat produced may be ignored. Five variables now remain, of which three -- water evaporating surface, superheating surface and grate area are measures of the size of the boiler. If these three can be combined into one variable, a single value giving an index of the capacity of the boiler, the number of variables will be reduced to three. These are the firing rate, the boiler "index", and the tube length.

The index number referred to must be some figure made up by a method of adding the evaporative capacity of the component parts of the boiler: the firebox surface, the tube heating surface, and the superheating surface. Since for the usual type of firebox the relation between grate area and firebox heating surface is almost constant, the grate may be used in place of the heating surface as a measure; this is a convenience since in some cases the information as to the grate area might be available, there being none as to firebox heating surface. For similar reasons, it is convenient to use the total heating surface in place of that of the tubes.

In the case of the Coatesville Boiler Tests we have a set of carefully conducted tests upon a typical modern boiler, partitioned into two sections, in order that the evaporation of the tubes and of the firebox surface might be measured separately. No other recent data is available on this subject, therefore the conclusions of the Coatesville tests will be adopted. Appendix VII-C gives the results of these tests in abbreviated form*, and shows the ratios between the evaporative capacities of the boiler as a whole and of the firebox. For coal rates which are of interest in locomotive service, the average ratio is 4.78: that is, the evaporation per square foot of firebox heating surface is nearly five times as much as that of the total heating surface, under service conditions. It being desired to find the relation between the evaporative capacity of the grate and that of the total heating surface, it is necessary to investigate the relation between the grate and the firebox surface. To accomplish this, the data for some seventy modern engines was plotted as shown in Appendix VII-D with the conclusion there noted, that is, that for current practice, the figure 3.75 is representative of the relation between grate area and firebox surface. Accepting these figures, one square foot of grate is more valuable as an evaporative agent than a square foot of the total heating surface in the ratio of 3.75 x 4.78 to 1, or 18 to 1. Hence, in a composite figure representing the capacity of the boiler as a whole, the grate area must be

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* See the report of the tests, published by the Jacobs-Schupert United States Firebox Co.

given 18 times as much weight as the heating surface.

The boiler of any locomotive may be conceived to be made up entirely of directing heating surface -- that is, a long grate might occupy the whole of the bottom part of the boiler, no tubes being used. If locomotives were actually built in this way, the comparison of boiler capacities would be a simple matter: with the use of such an equivalency figure as has been developed in the preceeding paragraph, the selection of a figure representing the "equivalent" size of the boiler becomes equally simple. This figure has been termed the "Equivalent Grate" or "E.G." in the further discussion, and for a locomotive using saturated steam

$$E.G. = \frac{\text{Heating Surface*}}{18} + \text{Grate Area}$$

In a discussion of the evaporation of any boiler the effect of the addition of a superheater is not at first apparent, -- the superheating surface has not in general been considered as evaporating surface, and the preliminary investigations on the basis of the equivalent grate made no use of it. However, the consistently higher evaporation per square foot of heating surface -- or per square foot of equivalent grate (when obtained as given above) -- in the case of superheater engines led to the conclusion that it was improper to ignore the heat transmitting value of the superheater tubes. The

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* In all discussions involving heating surface, the water side of the tubes has been used, this being the figure always given by the builders.

bulletins of the Pennsylvania Test Department show the evaporative value of a square foot of the superheater surface is from one-fourth to one-half of that of a square foot of the boiler considered as a whole, the value under ordinary service conditions approaching the latter figure. There is an incidental feature also in the case of the superheater engine: the large superheater flues have a higher evaporative capacity per unit area than that of the tubes on account of the higher temperature which prevails within them. It does not appear to be possible to evaluate either of the effects mentioned, but on account of its harmonizing effect upon the data which will be shown later, the assumption has been made that on account of both the indirect and direct effects of the presence of the superheater, its surface should be added to that of the tubes and firebox and considered a part of the evaporative surface. Thus for superheater engines the expression for the equivalent grate becomes

$$\text{E.G.} = \frac{\text{Heating Surface} + \text{Superheating Surface}}{18} + \text{Grate Area}$$

4. Comparison of Data on Equivalent Grate Basis. In comparing the efficiency of different boilers it is obviously unfair to make the comparison except at the same rates of combustion. Assume for example a locomotive boiler having a total heating surface of 2000 square feet and a grate area of 25 square feet; assume another with a total heating surface of 2500 and 50 square feet of grate. What is the "same rate of combustion" for these boilers? It is plain that it is not the same total

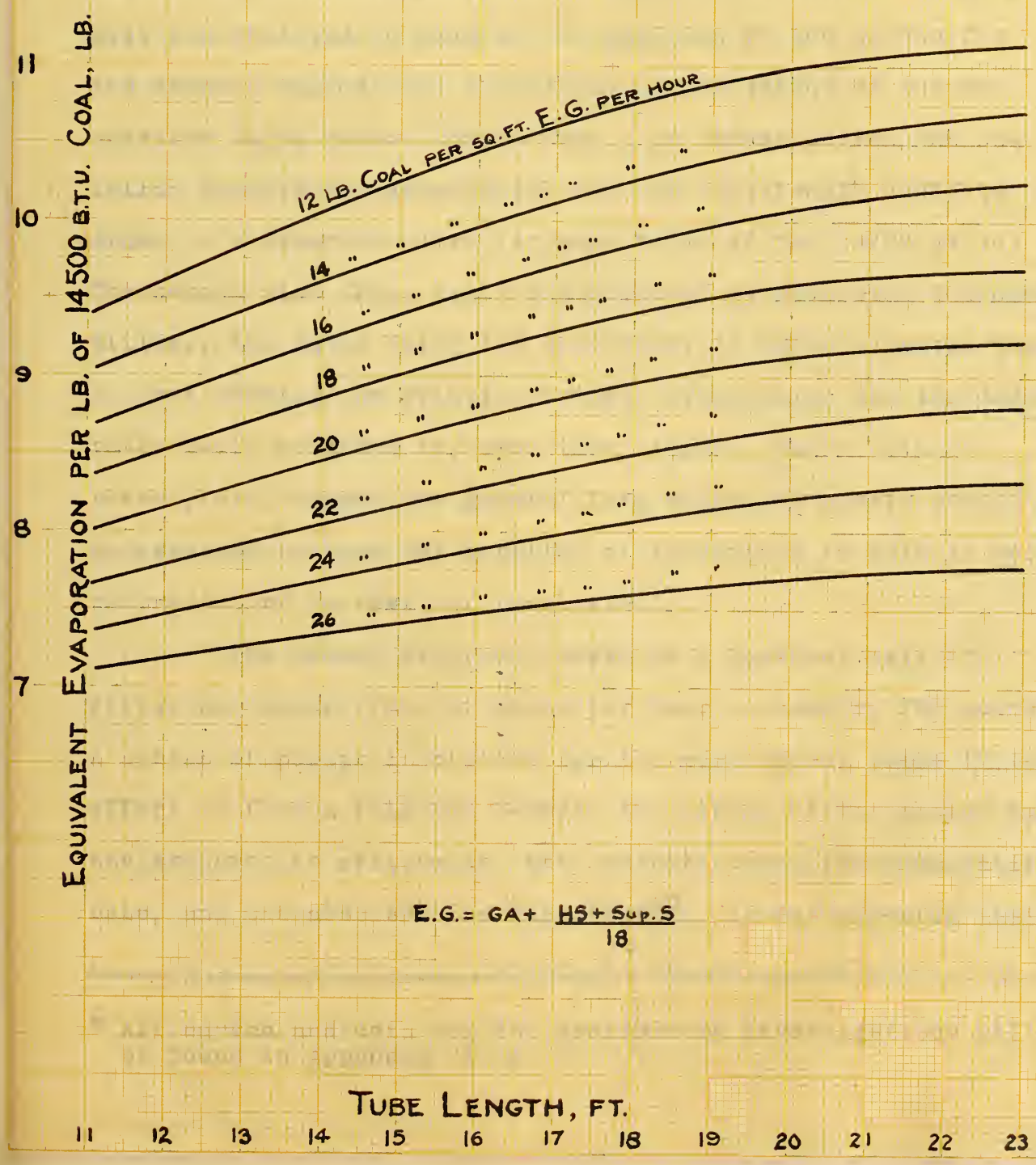
amount of coal fired per hour, since one boiler is larger in every dimension than the other. If the same amount of coal is burned per square foot of heating surface per hour -- say two pounds, coal will be fired upon the smaller grate at the rate of 160 pounds per square foot per hour, and upon the larger grate at the rate of 100 pounds per square foot per hour. In this case the efficiencies of the heating surfaces might not be greatly different, but that of the furnaces would not be comparable. Exactly the opposite situation is met when the same amount of coal is fired per square foot of grate surface per hour -- approximately equal efficiency of the grate and firebox, but much variation in that of the heating surface. Suppose 100 pounds per square foot of grate were fired in the case mentioned: the smaller boiler would be burning coal at the rate of 1.25 pounds per hour per square foot of heating surface; the latter at the rate of 2 pounds per hour. Evidently the rate of firing must be figured on some composite basis -- since the same rate of combustion implies the same efficiency. The value for the "equivalent grate" previously developed provides this basis of comparison. By comparing boilers on the basis of a single index-figure representing their evaporative capacities the proper rate of firing on which a comparison of the several evaporations should be made can be stated in terms of this index.

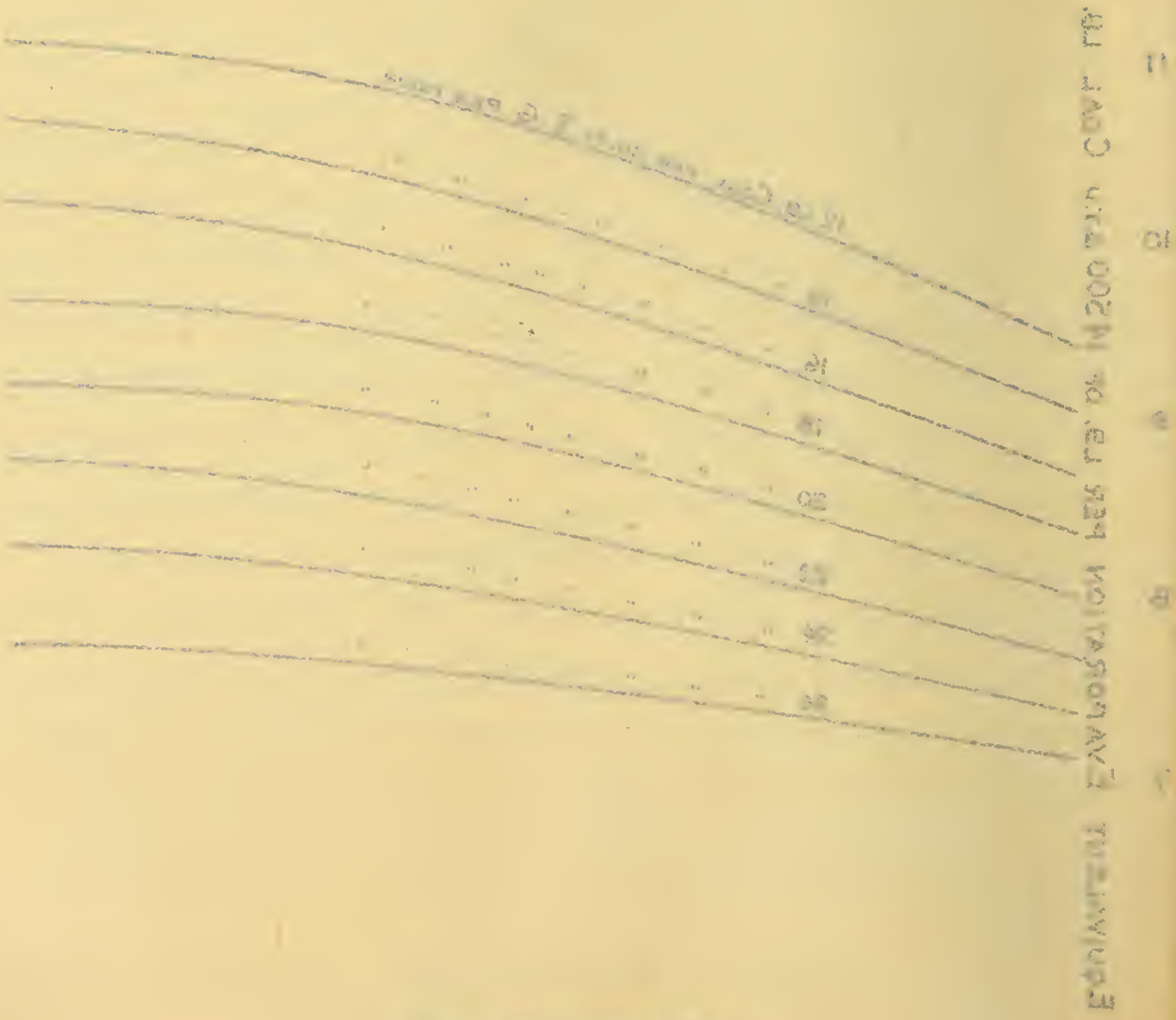
The combination of three of the five variables remaining in the problem (see p. 89) into a single variable reduces the number remaining to three. These are: the quantity of coal

fired, the index number representing boiler capacity (the equivalent grate), and the tube length. The first two of these are readily combined to give the rate of firing, and this ratio with the tube length remain as those which must be shown in the final plot, with the evaporation rate. A study of the relation of the equivalent grate figure to the coal fired shows that 18 pounds of coal per square foot of equivalent grate represents a reasonable rate of firing, and that the limits are perhaps twelve and twenty-five pounds for service conditions. From the available test data (the laboratory tests of twenty-seven engines) the required evaporation rates are computed for firing rates of fourteen, eighteen and twenty-two pounds of coal per square foot of equivalent grate per hour. The data as calculated and used is given in Appendix VII-E. The results are plotted in Fig. 20, with curves for 12, 16, 20, 24, and 26 pounds mathematically interpolated. The variation in evaporative performance of engines of almost identical proportions is considerable, but the relation between the averages is very consistent as shown by the grouping of the points in the original plots, shown also in Fig. A-9. The values given by the curves are the equivalent evaporation rates.

5. Preliminary Investigations. The curves of Fig. 20 are the outgrowth of a number of preliminary investigations which are directly connected with the various steps of the study of the problem as heretofore outlined. These preliminary investigations were as follows:

Fig. 20
Evaporation Curves




EVAPORATIVE TURBINE
EVAPORATION CURVES FOR 200 MW 34 BY 34 MOLYBDENUM
TUBE LENGTH, FT.
11
10
9
8
7
11
12
13
14
15
16
17
18
19
20
21
22
23

First, a firing rate of one pound of coal per square foot of heating surface per hour is assumed, and the relation between the evaporation rate, the ratio of heating surface to grate area, and the tube-length is studied.* In the first plot, (Fig. A-10) the tube-length was taken as the abscissa and the evaporation per pound of coal as the ordinate. This plot was expected to show by the position of the points for the several engines and the noting of the values of the respective GA/HS ratios that a family of curves giving the relation between the evaporation and the tube-length could be drawn -- a separate curve for each value of the GA/HS ratio. The second plot (Fig. A-11) was produced by reversing the conditions, the ratio being the abscissae, it being expected that a curve showing the relation between evaporation and the GA/HS ratio could be drawn for each tube length. While both of these plots suggest the general form which the curves should be expected to take the position of the points in both is too contradictory to admit of conclusions.

The second study was based on a constant rate of firing per square foot of grate per hour -- namely, 100 pounds. A series of points is plotted for the more recent tests in the effort to find a relation between the design of the boiler and the evaporative efficiency, the ordinate being the evaporation rate, and the abscissa the tube length. It was expected that

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* All curves and data for the preliminary investigations will be found in Appendix VII-F.

the position of the various points would indicate the nature of a family of curves as before, one curve for each ratio of GA/HS. The results are inconsistent, and apparently of no value. (See Fig. A-12).

The third investigation was an effort to find the law of the variation of the evaporation with the total heating surface, the grate remaining constant. Evaporation data is available for ten engines having grates varying in size from 53.3 to 55.8 square feet, the variation in total heating surface being from 2105 to 4619 square feet, not including superheating surface. The general trend of the points appears to be in the right direction -- showing a higher rate of evaporation (per pound of coal) for the high values of the HS/GA ratio, but the variation is too great to warrant the drawing of an average curve.

The fourth investigation was built upon the equivalent grate, not taking into account the superheating surface. In this study curves were plotted for various rates of firing between the evaporation rate and the tube-length (see Fig. A-14). Only the more recent test-data were used, and the agreement is very satisfactory, the relation between the average points giving curves of the shape and position expected. However, the omission of the St. Louis tests from this study is difficult to justify: while it is true that these tests show a uniformly low boiler efficiency compared with the later tests, the evidence shows that they were properly conducted, and should be included, even though they are discordant and make generalization more difficult.

The fifth investigation acted on this conclusion, including all available data. A little better concordance was secured by rearranging the plot -- that is the evaporation per pound of coal was plotted in place of that per square foot of heating surface. The relation between the average points for each tube length is quite satisfactory (shown in Fig. A-15), though the variation in the individual points for each tube-length is wide enough to vitiate the conclusions. At this point it becomes evident that the superheater boilers are of much greater capacity and efficiency per square foot of water-evaporating-surface than those without the superheater, therefore in the next investigation the formula for the equivalent grate was modified.

No plot is made for this sixth investigation. The rate of firing was taken at eighteen pounds of coal per square foot of equivalent grate per hour, and the superheater was taken into account in figuring the equivalent grate as follows:

$$EG = G.A. + \frac{H.S.}{18} + \frac{Sup.\ S}{50}$$

thus giving the superheating surface one-third of the weight of the evaporating surface. The data produced showing that the evaporation of the superheater boilers is still materially better than those without the superheater, the seventh investigation was built up by including the superheating surface with the evaporating surface thus:

$$EG = G.A. + \frac{H.S. + Sup.\ S.}{18}$$

This investigation has been previously discussed. In the agreement of the component points it is more satisfactory than any of the others.

6. Method of Using Curves. Fig. 20 shows the curves for the final investigation redrawn, and these curves furnish a method of estimating evaporative performance, giving consideration to all of the major variables. As an example of the use of the curves, assume a boiler of the following proportions:

Total heating surface, 3000: superheating surface, 720; grate area, 54, tube-length 16 feet. Rate of firing, 4500 pounds of coal per hour.

The equivalent grate is 54 + (3000 + 720)/18 = 260.6, therefore the rate of firing is 4500/260.6 = 17.3 pounds per square foot E.G. per hour. Interpolating for this firing rate on the abscissa representing 16 feet tube length, we have an evaporation of 9.5 pounds per pound of coal, or a total of 41600 pounds per hour, equivalent.

VIII. OUTLINE AND SUMMARY OF THE PROCESS.

1. Directions for Applying the Process.
   a. The data required.
   b. Method of finding the equivalent grate.
   c. Method of finding the coal-factor.
   d. Curves for estimating evaporation, and method of using.
   e. Determination of pull at 600 feet per minute piston speed.
   f. Determination of pull at other speeds.
2. Application of the Process.
   a. Superheated steam locomotives, actual and hypothetical cases.
   b. Saturated steam locomotives, actual cases.

1. The Process Summarized. To make an accurate application of the process for estimating the speed-pull relation for any engine as outlined in the preceding pages a considerable amount of information is required, but there is nothing needed that would not be available to anyone desirous of predicting the performance for location purposes if the engines are already in use, or even designed. For rating calculations, all of the figures required will be in the possession of the mechanical department. For the purposes of design, the process in itself furnishes all of the more

important dimensions (or at least the relations between them) providing the service requirements are known.

The necessary data are as follows:

Quantity and quality of coal to be fired per hour, or the rate of firing to be maintained during the period when the engine is to give the required output.

Total heating surface, grate area, and tube length.

Cylinder size, boiler pressure, and driver diameter.

Whether or not the engine is to use superheated steam.

If an estimate of the drawbar pull is desired, as is usually the case, it will be necessary to know in addition the weights of the locomotive and the tender in order that the friction and resistance may be calculated.

The first step in the process is to find the value of the equivalent grate, which is a rough measure of the boiler capacity of the engine under consideration. This is done by adding to the actual grate area one-eighteenth of the heating surface. For the heating surface, the figures as given by the builders should be taken: that is, water-side of the tubes, and fire-side of the firebox. The superheating surface is also added in, for reasons previously stated, so that expressed as formula,

$$EG = \frac{HS + SHS}{18} + GA$$

Knowing the rate of firing and the tube-length, next find from Fig. 21 the evaporation per pound of coal. This estimate is based on coal of an average heating value of 14,500 B.t.u., therefore some factor must be applied to make the estimate applicable to the quality of the coal to be used or in

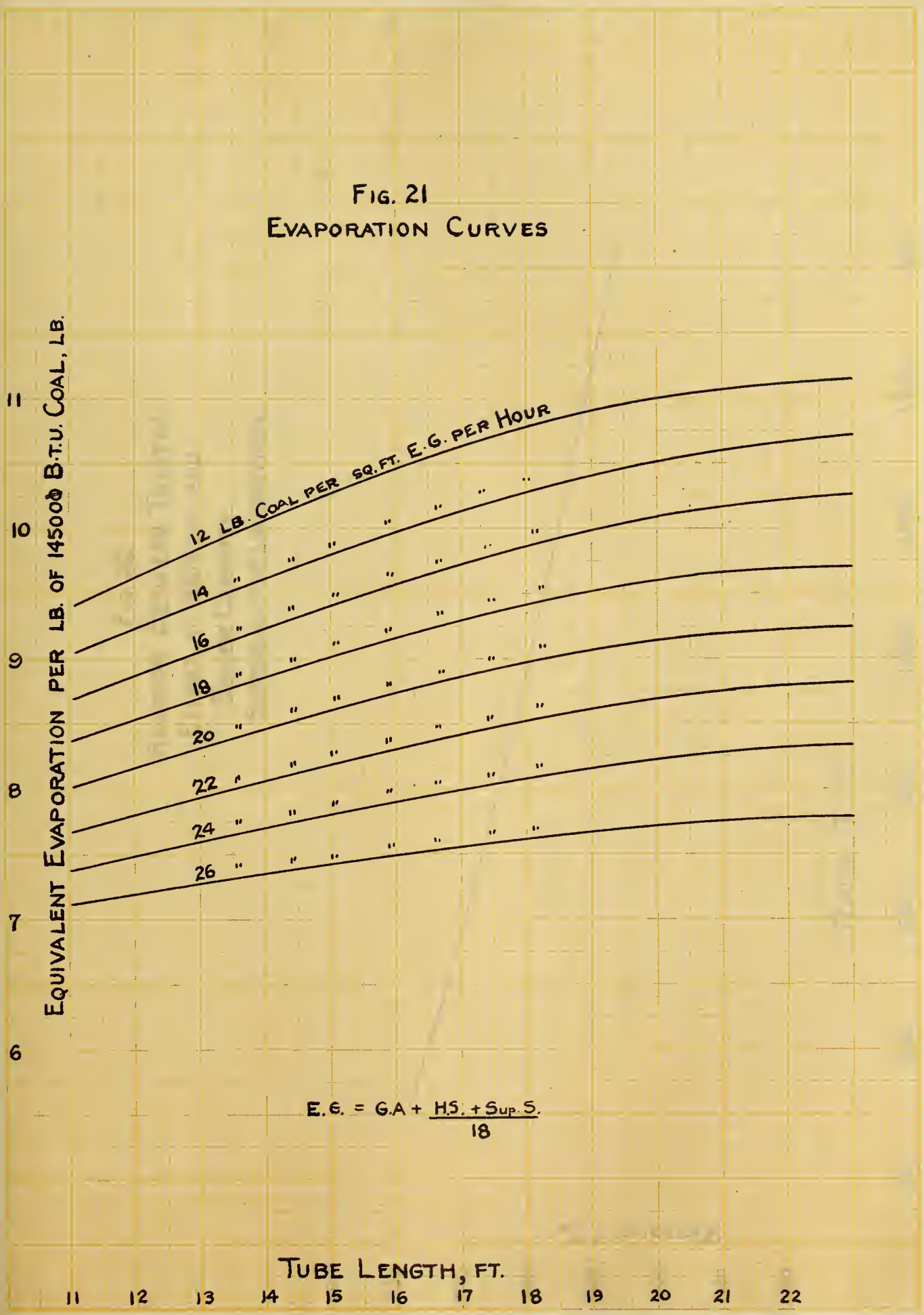


Fig. 21
Evaporation Curves
Equivalent Evaporation per lb. of 14500 B.T.U. Coal, lb.
11
10
9
8
7
6
12 lb. Coal per sq. ft. E.G. per Hour
14 " " " " " " "
16 " " " " " " "
18 " " " " " " "
20 " " " " " " "
22 " " " " " " "
24 " " " " " " "
26 " " " " " " "
E.G. = G.A + H.S. + Sup. S. / 18
Tube Length, ft.
11 12 13 14 15 16 17 18 19 20 21 22

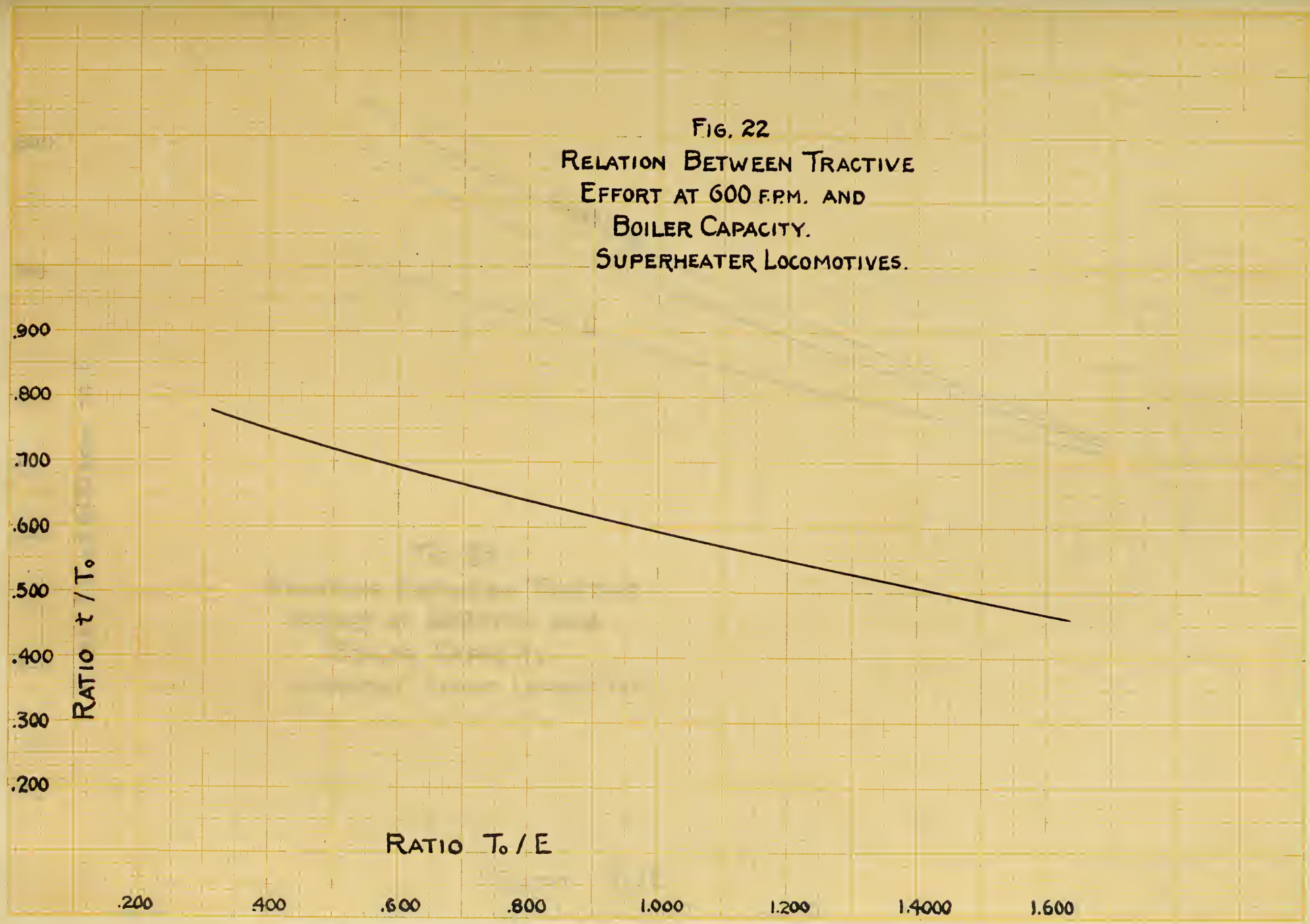


Fig. 22
Relation Between Tractive
Effort at 600 F.P.M. and
Boiler Capacity.
Superheater Locomotives.
.900
.800
.700
.600
.500
.400
.300
.200
Ratio t / To
Ratio To / E
.200
.400
.600
.800
1.000
1.200
1.4000
1.600

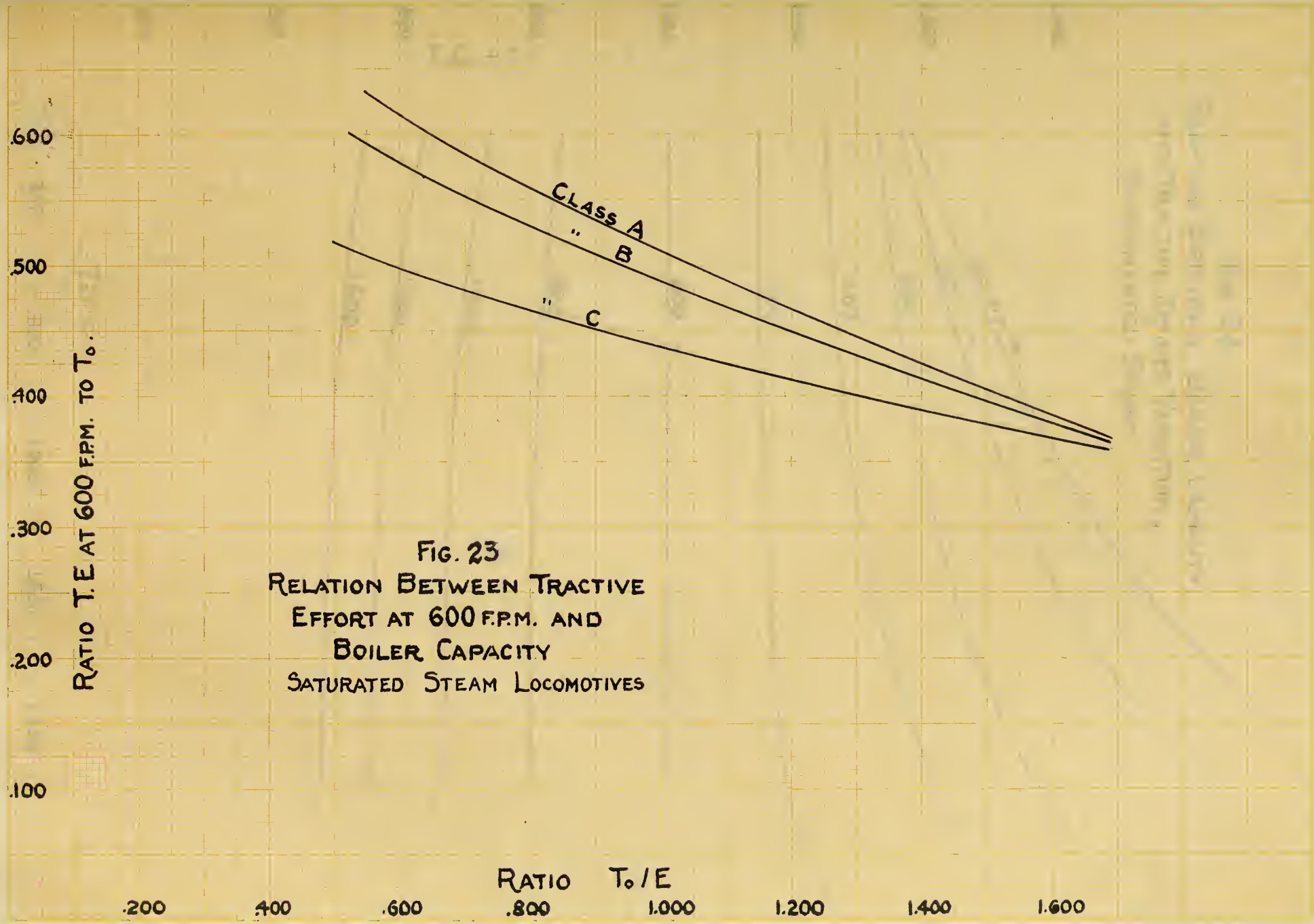
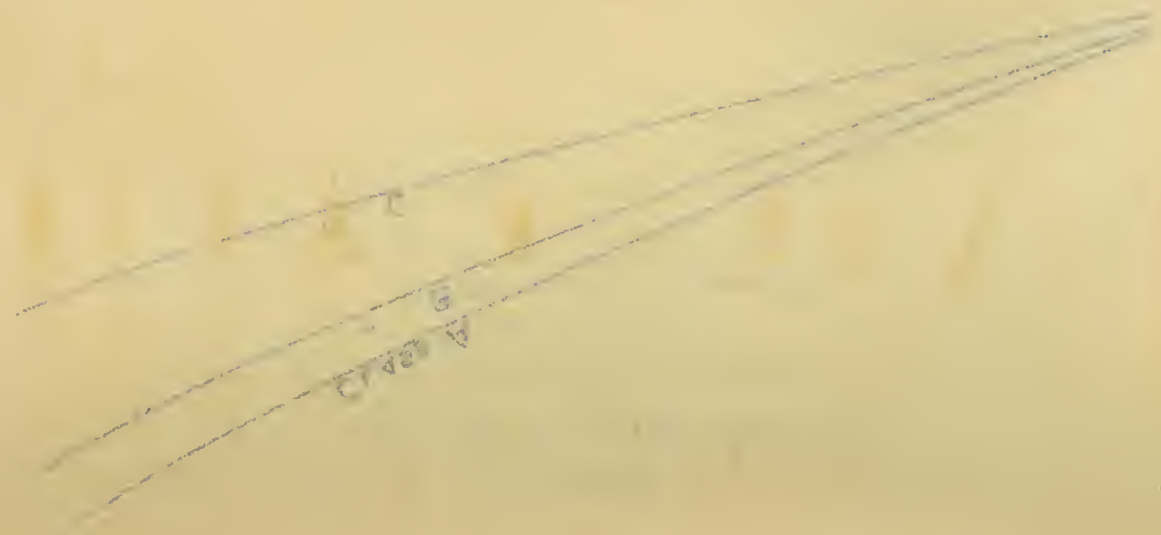


Fig. 23
Relation Between Tractive Effort at 600 F.P.M. and Boiler Capacity
Saturated Steam Locomotives


CURVE A
B
C

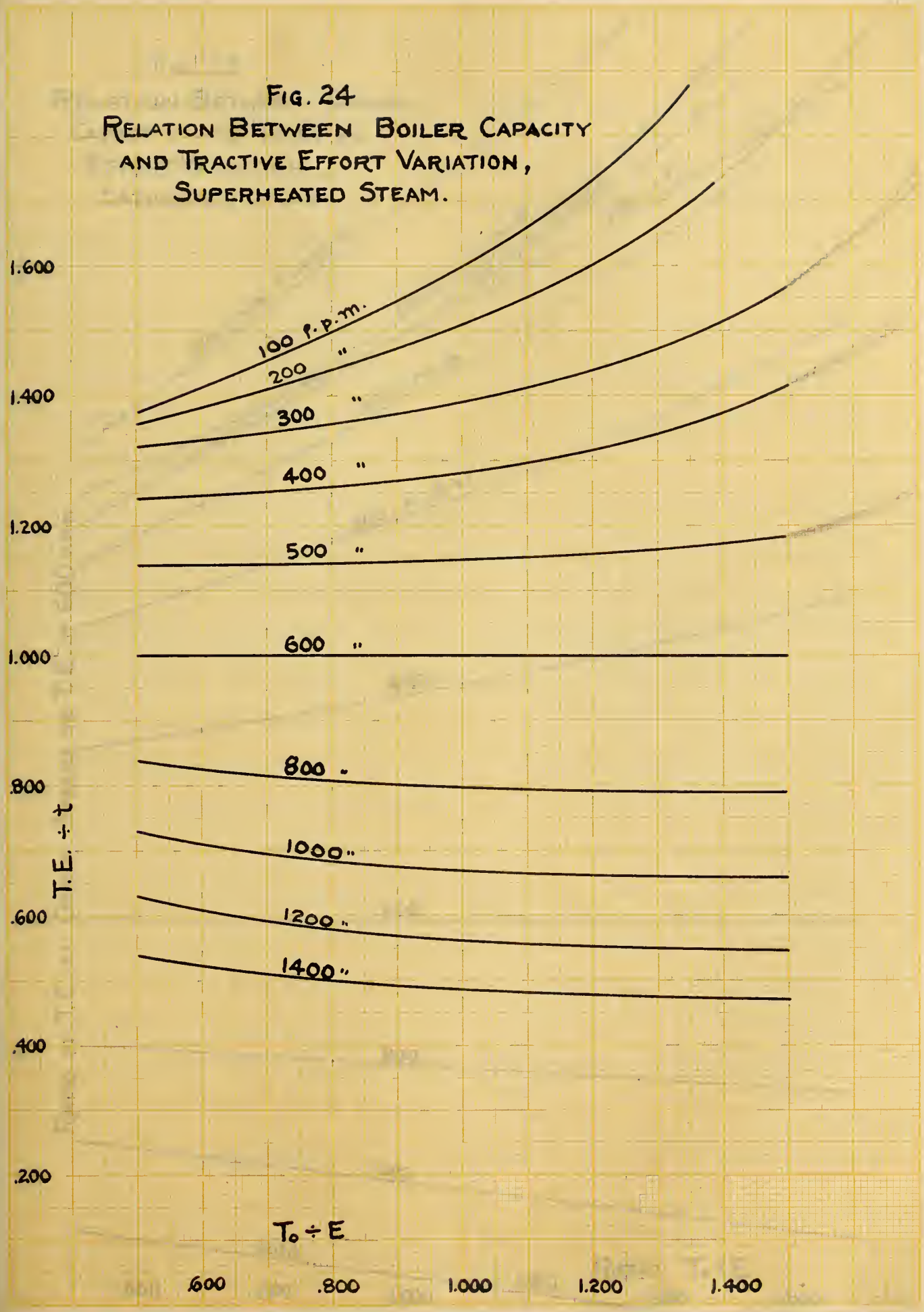


Fig. 24
Relation Between Boiler Capacity and Tractive Effort Variation, Superheated Steam.

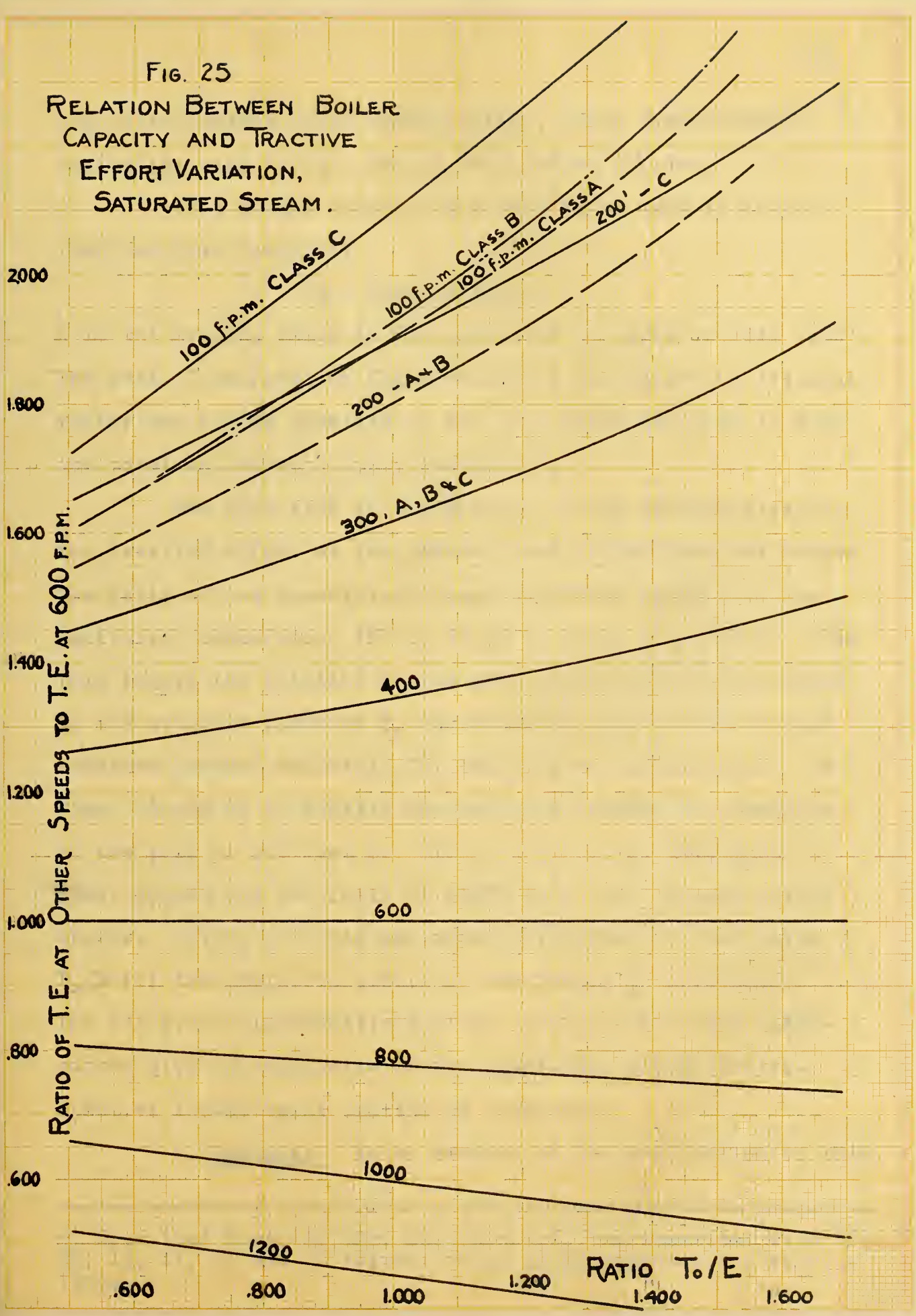


FIG. 25
RELATION BETWEEN BOILER CAPACITY AND TRACTIVE EFFORT VARIATION, SATURATED STEAM.

use on the engine under investigation. This correction is called the coal factor, and is obtained as follows:

If X is the value of the coal to be used in B.t.u., then the coal factor is

$$(X - 3000)/(11500)$$

X is the heating value of the coal used in terms of "dry coal". The rate of evaporation found should be multiplied by the coal factor and by the quantity of dry coal fired per hour to find the total equivalent hourly evaporation.

The next step in the process is the determination of the tractive effort at the piston speed of 600 feet per minute. The ratio of the theoretical tractive effort (pld^2/D) to the equivalent evaporation (E) is shown in Figs. 22 or 24.* From this figure the relation of the pull at 600 feet piston speed to the quantity pld^2/D or T_0 may be found, and the value thus obtained becomes the basis for the further calculations. In Figs. 23 and 25 is plotted the relation between the fraction of the pull at 600 feet per minute that can be developed at other speeds and the ratio of pld^2/D to E, for various piston speeds. Select the abscissa which corresponds to the ratio T_0/E for the engine in question, and follow it vertically. The successive intersections of the abscissa with the speed-curves give the ordinates of the speed-pull curve, the respective speeds being plotted as abscissae.

2. Examples. As an example of the application of this

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* Note that Figs. 21, 22, 23, 24 and 25 are identical with Figs. 20, 13, 17, 14 and 18 respectively, being repeated for convenience.

process, it will be carried out for a case in which full data are available, the tests of engine 999, a superheater, Mountain type engine on the Chicago, Rock Island and Pacific. The dimensions are as follows:

Cylinders	28 x 28"
Drivers	69"
Boiler Pressure	185 pounds
Wt. Total Engine	166 tons
Wt. on Drivers	112 tons
Wt. on Trucks	54 tons
Wt. on Tender	80 tons
Heating Surface	4117 sq. ft.
Superheating Surface .	944 sq. ft.
Grate Area, ...	62.7 sq. ft.
Tube length	22 ft.

The maximum coal fired per hour was 129.4 pounds per square foot of grate or a total of 8110 pounds. Its B.t.u. value was 13100, hence the "coal factor" is .88. The equivalent grate is 344.7 square feet, and the rate of firing 23.5 pounds of coal per square foot of equivalent grate per hour. From Fig. 21 the corresponding evaporation for 14500 B.t.u. coal should be 8.4 pounds of water per pound of coal, or 7.4 pounds for 13100 B.t.u. coal. Dividing this by the factor of evaporation 1.3, we have 5.68, and multiplying by the amount of coal, the total actual evaporation as estimated is 46,000 pounds. From the tests the average evaporation was 46,400 pounds, a very satisfactory check. The coal and water rates as used represent the average over a run of 140 miles with a very heavy train (1000 tons of steel passenger equipment), a constant grade of 0.8% over the most of the section, and only five stops*

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* The information concerning water and coal rates, etc., is derived from a personal letter from Mr. H. A. Houston, Assistant to the General Mechanical Superintendent.

so that the average rates are fair representations of the engine's performance. However, this engine delivered a little over 3000 indicated horsepower at its best, and with an evaporation of 46,400 pounds, this would give a water rate of 15.5 pounds per I.H.P.H., better by one pound than the average of the five best engines tested upon the Altoona plant at the same piston speed. It is obvious that the average evaporation must have been exceeded at the highest rates of working. As the conditions of the test were favorable in the highest degree to an economical performance and the engine was skillfully handled, a water rate of 17 pounds (corresponding to that developed by the two least economical of the five Pennsylvania engines mentioned) seems reasonable to base further estimates upon, since the highest rates of firing can not be learned. This corresponds to a ten per cent. increase in the evaporation, that is, 51,000 pounds actual, or 66,400 pounds equivalent evaporation.

For this engine pld^2/D is 58,600 pounds, and E is 66,400, therefore $T_o/E = .885$; the value of the ratio t/T_o corresponding is .62, from Fig. 22 the cylinder tractive effort at 600 feet per minute piston speed is .62 x 58,600 or 36,400 pounds. For other speeds, following vertically the abscissa-value .885, we have:

Piston Speed	Percent of T_{600}	Tractive Effort.	M.P.H.
100	153	55400	4.4
200	147	53500	8.8
300	137.5	49500	13.2
400	126	46000	17.6
500	115	42000	22.0
600	100	36400	26.4
800	80	29100	35.2
1000	67.6	24600	44.0
1200	57.5	21000	52.8
1400	49.8	18200	61.5

The builders of this locomotive published in the Railway Age Gazette (several numbers in 1915) an advertizement giving data for the drawbar-pull curve for this engine. The figures reduced to piston speed and corrected for friction and resistance are reproduced herewith. The friction correction corresponds with that found for a Consolidation locomotive of similar weight on the drivers tested by the Pennsylvania at Altoona. The resistance is calculated as previously described.

Speed-pull Curve for Engine 999, C.R.I. & P.

Speed in MPH	Piston Speed ft.per min.	Draw-bar Pull	Friction plus Resistance	Cylinder Tractive Effort
11.6	263	45700	4600	50300
15.5	352	42500	4600	47100
22.1	504	39000	4600	43500
27.0	614	32400	4600	37000
30.0	698	29300	4600	33900
35.6	809	25000	4700	29700
40.1	910	21900	4700	26600
45.0	1022	19200	4800	24000
49.9	1132	17900	4900	22800
54.9	1248	15600	5000	20600

The calculated and actual curves are compared in Fig. 26. It will be noted from an inspection of this figure that the agreement at speeds above 15 m.p.h. is practically perfect, and at

lower speeds the calculated pulls are well within the capacity of the engine. If a curve were drawn through the points, the calculated starting tractive effort would exceed that actually developed by about 2000 pounds.

As a further instance of the application of the process, it will be used to determine the speed-pull curve of the hypothetical engine used in investigating the A. R. E. A. method, in Section III. The data required are as follows:-

Type	Consolidation
Cylinders	24 x 30"
Drivers	60"
Boiler Pressure ...	200 lb.
Heating Surface ...	3244 sq. ft.
Superheating Surface	750 sq. ft.
Grate Area	65 sq. ft.
Tube length	15.5 ft.
Coal used	13000 B.t.u.
Rated tractive effort	48900 lb.
(85% mean effective pressure)	

For this engine the coal-factor is .87. The equivalent grate is 276 square feet. A rate of firing of 6000 pounds of coal per hour will be assumed: the coal per square foot of equivalent grate per hour is 21.7.

From Fig. 21 the evaporation for this engine if 14,500 B.t.u. coal were used would be 8.55 pounds of water (equivalent) per pound of coal. Applying the coal-factor for coal of 13,000 B.t.u., the evaporation estimate becomes 7.45. The total evaporation per hour is then 6000 x 7.45 or 44,600 pounds. From the dimensions of the engine we have T_o = 57,600 pounds, therefore T_o/E = 1.29 and from Fig. 22 the corresponding value of t/T_o is .53. From this figure, the cylinder tractive effort at 600 feet piston speed is 30,500 pounds. Then for other

speeds, following the ordinates corresponding to the abscissa 1.29 we have:

Piston Speed	Ordinate	Tractive Effort	M.P.H.
100	1.79	54600	3.57
200	1.64	50000	7.14
300	1.46	44600	10.71
400	1.33	40600	14.28
500	1.16	35400	17.75
600	1.00	30500	21.40
800	.79	24100	28.54
1000	.66	20200	35.50
1200	.55	16800	42.60
1400	.47	14400	50.00

This curve as well as those for the 999 will be found plotted on the following page.

As an example of the application of the process to a saturated steam locomotive, the speed-pull curve will be calculated for engine 1028 of the Atchison, Topeka, and Santa Fe Railway.* This engine evaporated 22,400 pounds of water per hour (actual, throughout the duration of the trip), or an equivalent evaporation of 27,000 pounds per hour. The evaporation per hour of running time was 34,500 pounds (equivalent). A reasonable assumption is that half as much water was evaporated during the running time as during the time of standing, as none of the stops were long. Then the average of the two evaporations, or 30,800 pounds may be assumed as the working evaporation. The engine worked at uniformly high horsepower throughout the running time, therefore this evaporation is a

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* This data is available to the writer through the kindness of Mr. F. J. Prout, with the approval of Mr. MacFarland, Engineer of Tests.

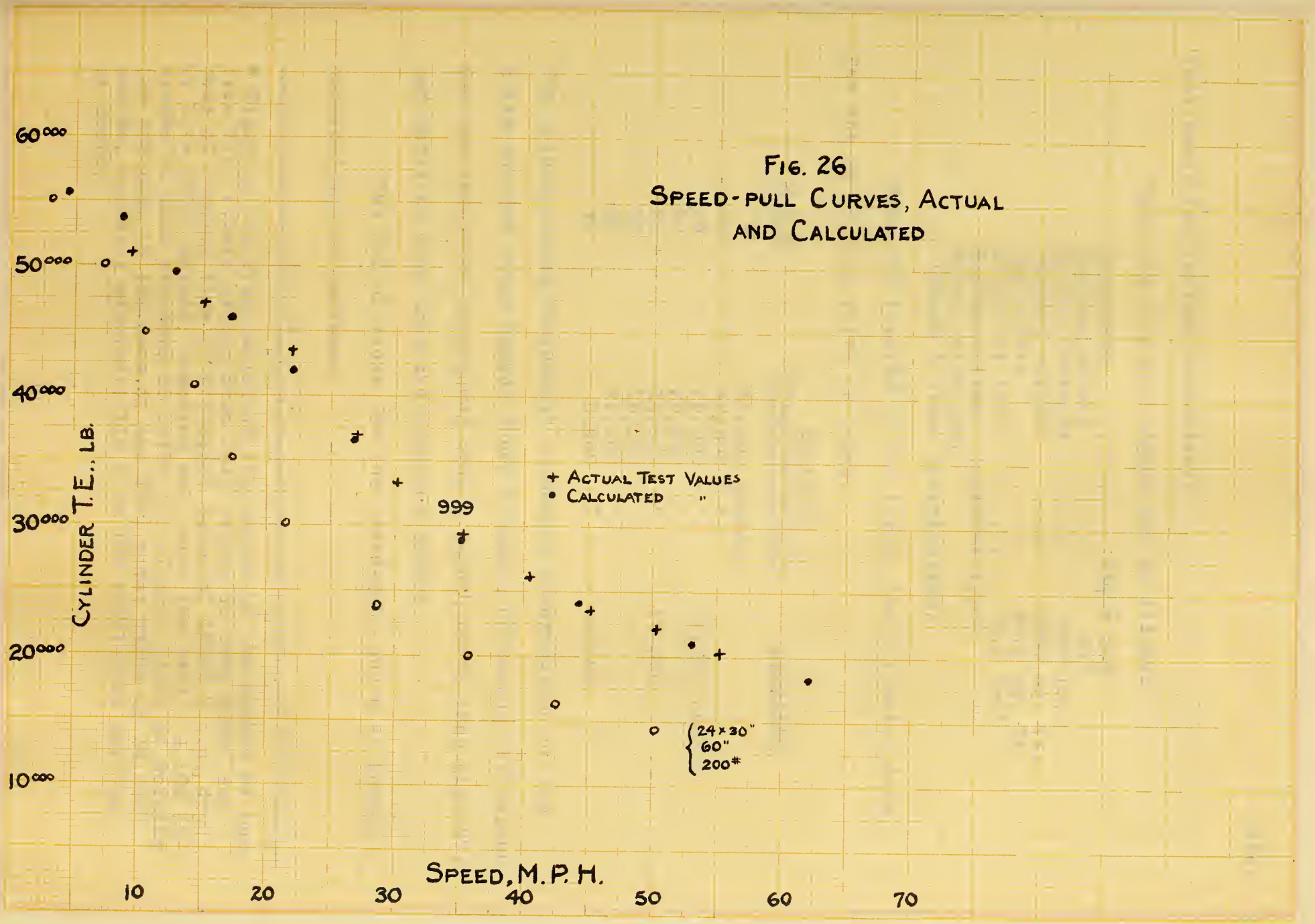

FIG. 26
SPEED-PULL CURVES, ACTUAL AND CALCULATED

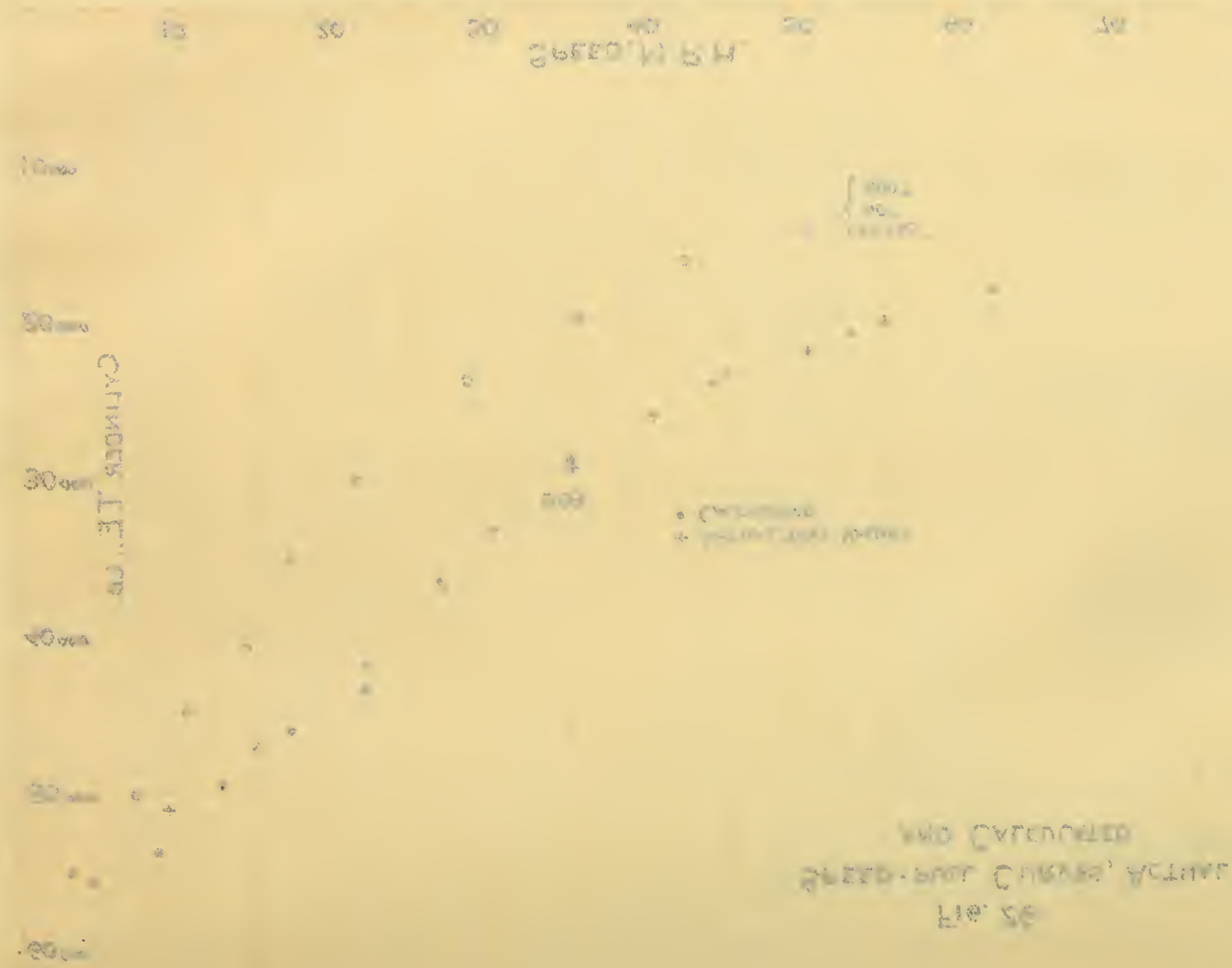
FIG. 56
SPEED-BOAT CHANGE' ACTUAL
AND CALCULATED
SPEED IN R.P.M.
CHANGE IN LIFT
10mm
20mm
30mm
40mm
50mm
60mm
10
20
30
40
50
60
70

fair basis for further estimates.*

The data for this engine are as follows:

Cylinders	$23\frac{1}{2}$ x 28"
Driving wheels	69"
Boiler Pressure	200 lb.
Heating surface	3560 sq. ft.
Grate area	53.5 sq. ft.
Length of tubes	19.0 feet
Piston-valves (Allfree type)	
Baker valve gear (Class A steam distribution).	

From the reports of the tests, the following data for the speed pull curve is taken:

M.P.H.	Cylinder Tractive Effort	Adjusted
6	37,400 pounds	
10	32,200 "	
16	25,000 "	26,200 pounds
19	23,200 "	
26	14,850 "	17,600 "
30	15,150 "	
35	11,600 "	
40	8,050 "	9,200 "

The column marked "adjusted" is merely a correction of the pulls for the three speeds shown by means of taking the values for the indicated horsepower from a smooth curve (See Fig. 27) and deriving from them the tractive effort.

The calculations for the speed-pull curve as herein outlined are as follows:

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* Figuring the evaporation of the engine by the method of Section VII, (there having been 3926 pounds of coal fired per hour of running time of a thermal value of 12000 B.t.u., dry) we have 27500 pounds, checking well with the test. A 10% increase of this estimate to cover the highest rates of working (as was made in the case of the 999) gives value of 30,000 pounds, agreeing substantially with the conclusion arrived at above.

For this engine pld^2/D or T_o is 44,800 pounds, and E is 30,800 pounds, therefore $T_o/E = 1.450$. From Fig. 23, for a Class A engine having $T_o/E = 1.450$, the tractive effort at 600 feet per minute piston speed is $0.415 \times T_o$, or 18,600 pounds. For other speeds:

Miles per hour	Piston Speed	Ordinate	Tractive Effort
4.05	100	2.27	42,300 pounds
8.10	200	2.14	39,800 "
12.15	300	1.80	33,500 "
16.20	400	1.44	26,800 "
24.30	600	1.00	18,600 "
32.40	800	.74	13,750 "
40.50	1000	.54	10,000 "

In Fig. 27 the calculated and actual curves are compared, and the agreement between the estimated and actual curves is most satisfactory at speeds above 15 m.p.h. Of the low speeds, more will be said later.

As a further example, the tractive effort curve will be calculated for Chicago and Northwestern engine 1015, an Atlantic type passenger locomotive, with the Young rotating valve. This will be treated as a Class B engine, as the rotating valve is an improvement of the same type as the piston valve. The other data* for this engine are:

Cylinders	20 x 26"
Driving Wheels	80"
Boiler pressure	200 lb.
Heating surface	3015 sq. ft.
Tubes, length	16 ft.
Grate area	46.3 sq. ft.

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* Data taken from the American Engineer and Railroad Journal, November, 1900, p. 333.

The calculations are as follows: the equivalent grate is 3015/18 + 46.3 = 213.0 feet. Illinois coal was fired at the rate of 6750 pounds per hour of running time, or 31.5 pounds of coal per hour per square foot of E.G. From Fig. 21, with this firing rate and a tube length of 16 feet, the evaporation per pound of coal is 6.3 pounds, extrapolating geometrically. This would give a total hourly evaporation of 42,500 pounds if 14,500 B.t.u. coal had been used. Assuming that the actual heating value of the dry coal was 13,000 B.t.u. the evaporation per hour is reduced to 37,000 pounds. Increasing this by ten per cent. as has been previously done* the value of E which will be used is 41,000 pounds.

For this engine T_o = 26,000 pounds, therefore T_o/E =.635. From Fig. 23, the pull at 600 feet piston speed is 0.570 x 26,000 pounds = 14,800 pounds. For other speeds:

Miles per hour	Piston Speed	Ordinate	Tractive Effort
5.5	100	1.72	25,400 pounds
11.0	200	1.61	23,800 "
16.5	300	1.49	22,000 "
22.0	400	1.28	18,850 "
33.0	600	1.00	14,800 "
44.0	800	.80	11,850 "
55.0	1000	.64	9,500 "
66.0	1200	.55	8,200 "

From the indicator cards taken during the test, the indicated horsepower curve has been plotted, and from this

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* The ten per cent. increase in the average evaporation which has been used is shown in this case to be a well grounded assumption. The average horsepower was 1250 and the normal maximum about 1400.

derived a curve for cylinder tractive effort. The maximum horsepowers attained at the various speeds were

16.6 M.P.H.	770 I.H.P.
30.0 "	1256 "
40.0 "	1383 "
50.0 "	1507 "
60.0 "	1434 "
70.0 "	1348 "

For the cylinder tractive effort curve, the data are as follows:

M.P.H.	I.H.P.	C.T.E.
20	1030	19,400 pounds
25	1160	17,300 "
30	1250	15,600 "
37.5	1360	13,600 "
45	1445	12,000 "
60	1435	9,000 "
75	1300	6,500 "

In addition to these figures, a starting card gives a mean effective pressure of 182 pounds, corresponding to a tractive effort of 23,700 pounds. In Fig. 28 the calculated and actual curves for this engine are compared, and it is seen that the agreement is very good at all but the lowest speeds.

It will be noticed that at low speeds the estimates are high for the curves of Figs. 27 and 28. In the case of both engines the estimate calls for practically 95 per cent of the theoretical rating at the start. While the data from which the 100 and 200 feet per minute curves are plotted in Fig. 25 are not satisfactorily concordant, yet this seems to be a reasonable estimate, and probably either of the engines could have delivered a larger starting pull than the test records indicate.

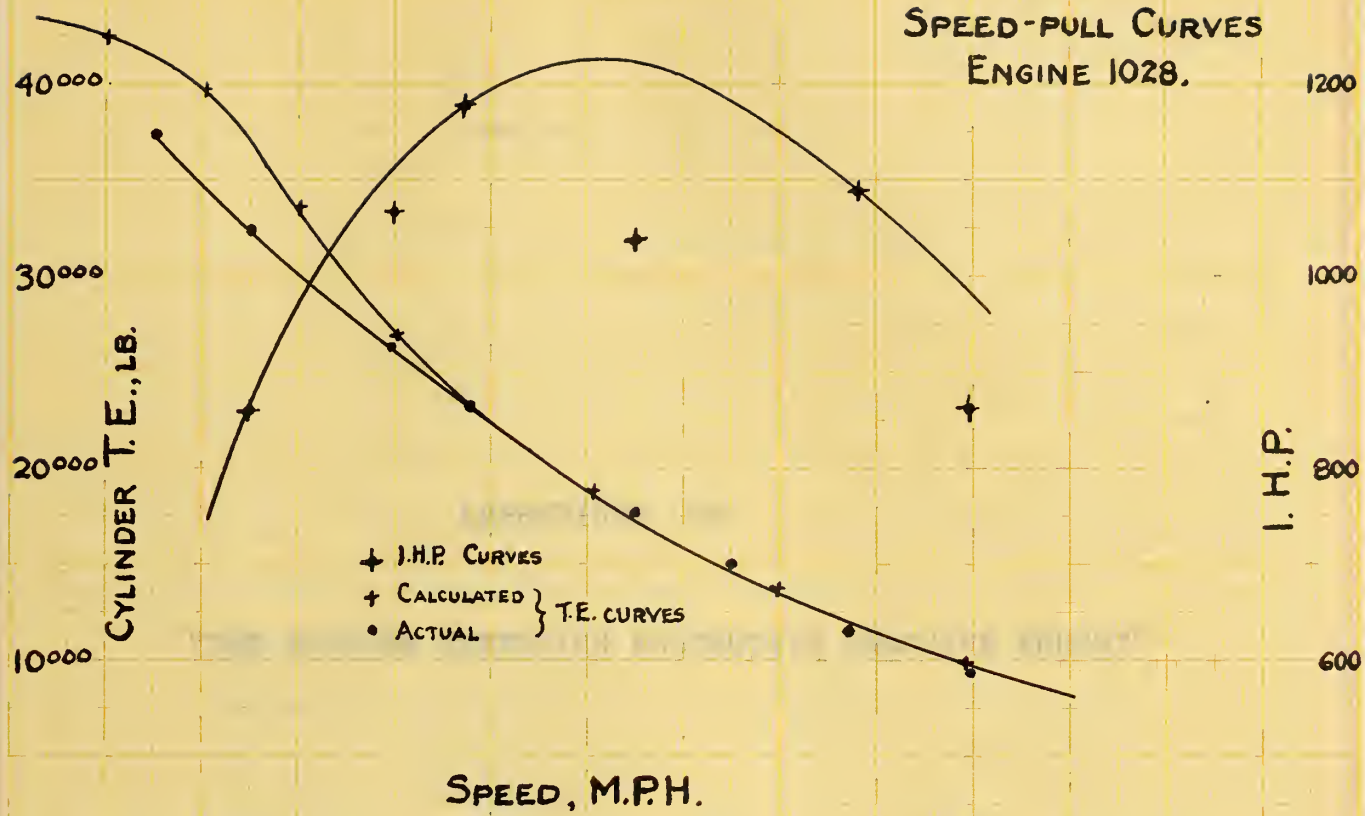


Fig. 27
Speed-pull Curves
Engine 1028.

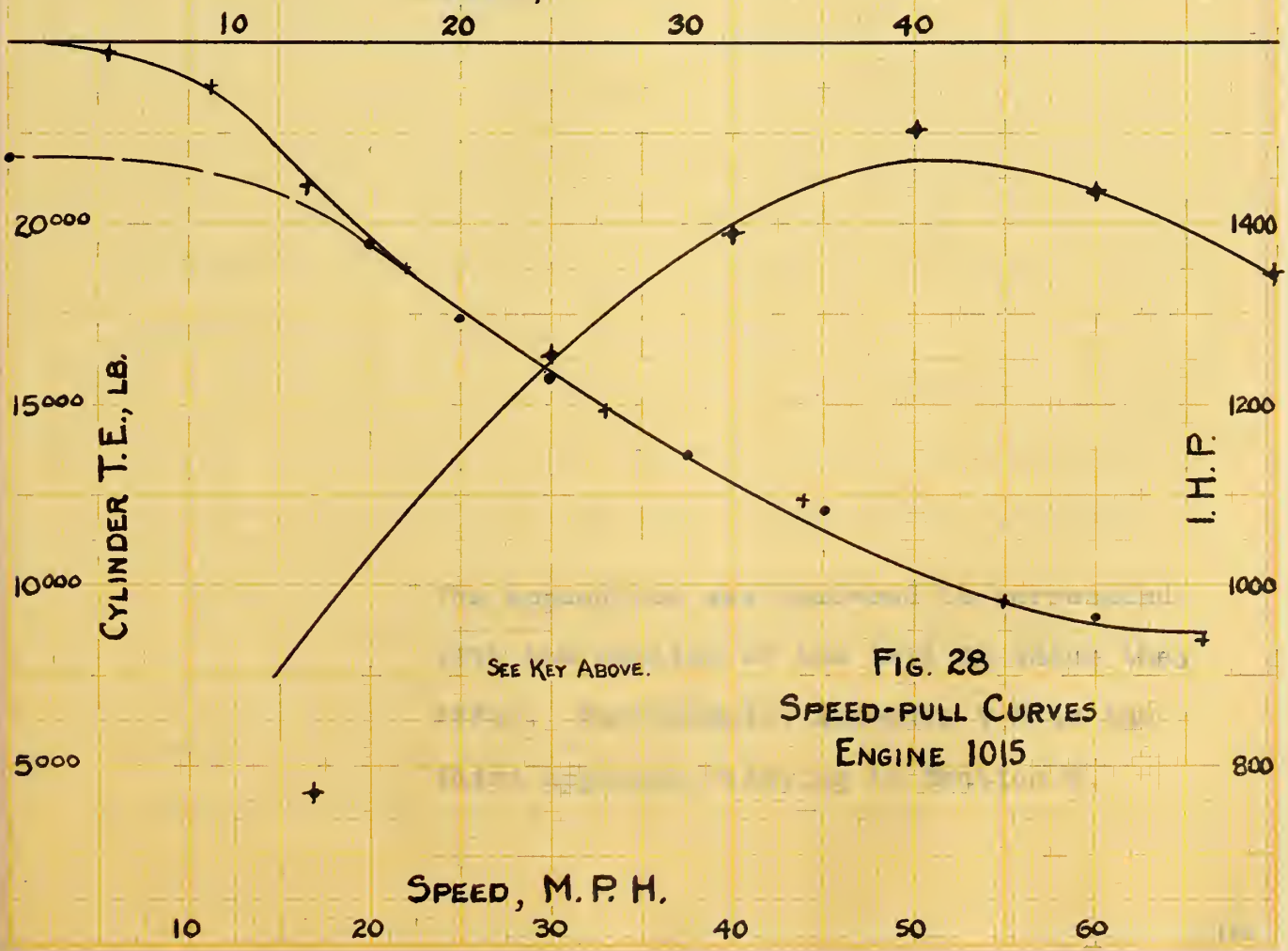
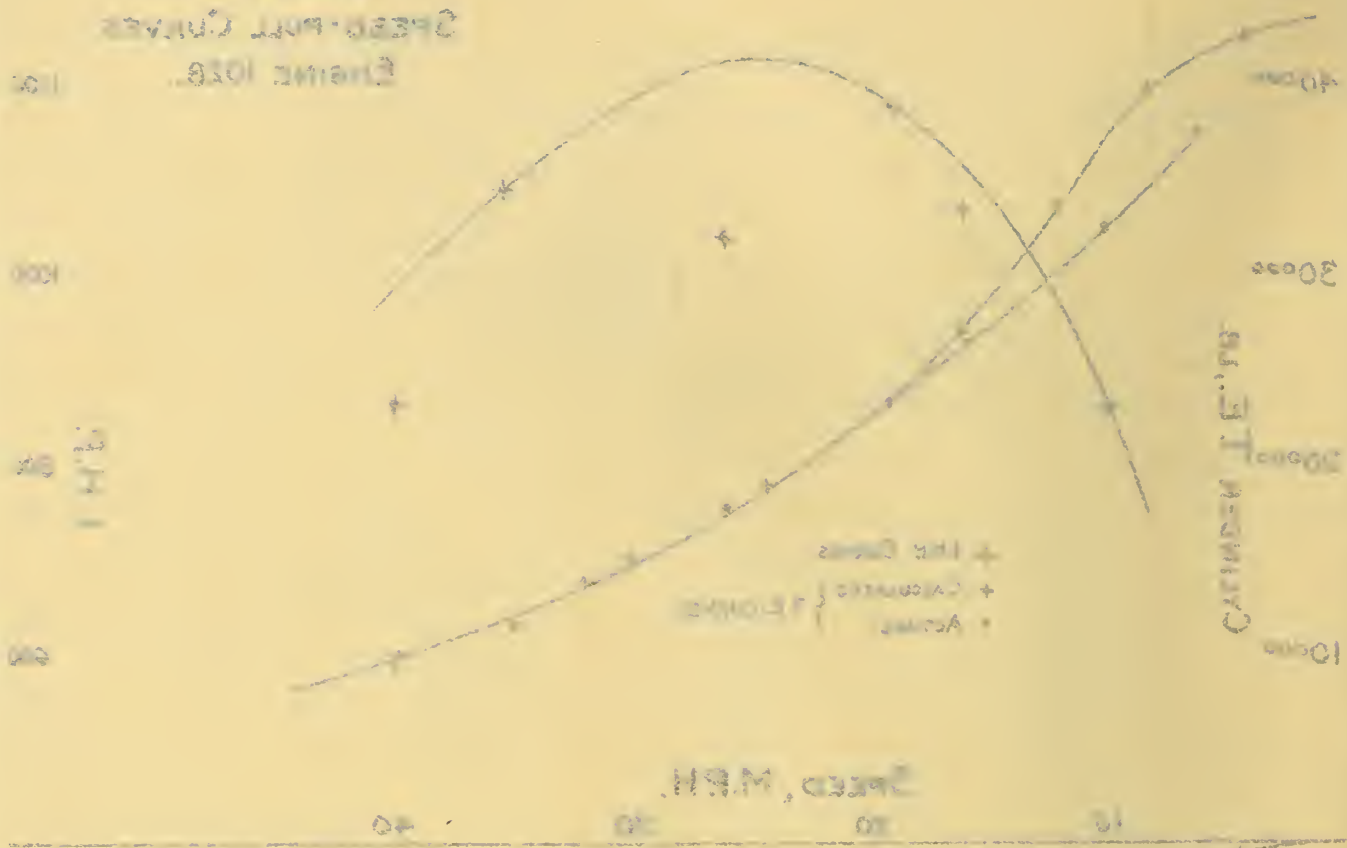
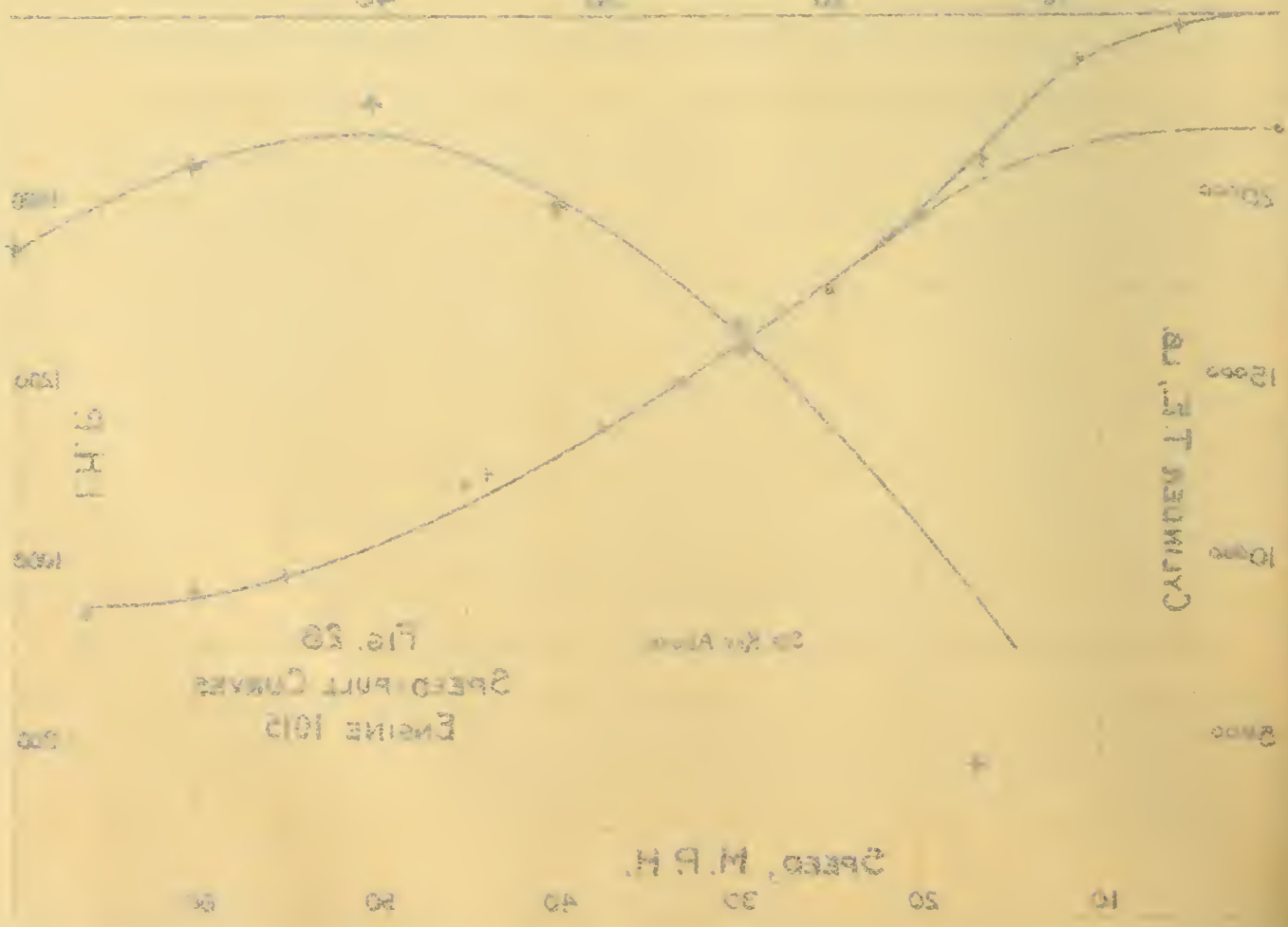


Fig. 28
Speed-pull Curves
Engine 1015


Speed, M.P.H.
Capacity, I.E., lb.
40000
30000
20000
10000
40
30
20
10


Speed, M.P.H.
Capacity, I.E., lb.
20000
15000
10000
5000
10
20
30
40
50

APPENDICES TO

"THE FACTORS AFFECTING LOCOMOTIVE TRACTIVE EFFORT"

The appendices are numbered to correspond with the section of the text to which they refer. For example, Appendix V-C is the third appendix relating to Section V.

APPENDIX II-A.

COMPARISON OF SPEED-PULL CURVES.

To render the theoretical curves comparable to each other and to a true test curve, the known performance of two Pennsylvania engines will be taken as a basis for comparison. These engines are of the Consolidation type, and have the following dimensions in common:

Cylinders	24 x 28"
Diameter of drivers,	62"
Working pressure,	205 #
Weight on drivers,	105.5 tons
Weight on trucks, plus weight of tender,	92. tons
Grate area,	55.13 sq.ft.

One of the engines (the H8b) uses saturated steam, and has a water evaporating surface (water-side of tubes) of 3840 square feet. The other engine uses superheated steam, and has 3030 square feet of water evaporating surface, and 780 square feet of superheating surface.

The standard coal of the Pennsylvania Railroad averages about 14,000 B.t.u., and this value will be adopted where required in the various methods of estimating the speed-pull relation.

The data for the speed-pull curves for these engines, from dynamometer tests, is given in Table 1. Owing to the fact that drawbar-pull is given, a correction for machine friction and resistance is introduced. The friction

correction is based upon the actual friction developed by an almost identical engine in laboratory tests (see Bulletin 10 of the Pennsylvania Railroad Test Department), and the resistance of the trucks and tender is calculated from the tables given in Bulletin 43 of the University of Illinois Engineering Experiment Station.

TABLE 1, Data for Speed-pull Curves.

Engine H8b.

Speed, M.P.H.	Drawbar Pull	Correction	Cylinder T. E.
0	45,700	5,000	50,700 pounds
5	43,500	5,000	48,500 "
10	37,250	4,900	43,150 "
15	30,000	4,800	34,800 "
20	24,000	4,700	28,700 "
25	19,000	4,600	23,600 "
30	14,600	4,500	20,100 "

Engine H8sb.(Superheated steam.)

0	45,700	5,000	50,700 "
5	44,750	5,000	49,750 "
10	40,200	4,900	45,100 "
15	33,500	4,800	38,300 "
20	28,350	4,700	32,050 "
25	22,350	4,600	26,950 "
30	19,850	4,500	24,350 "

The figures given were furnished by Mr. O. C. Wright of the Pennsylvania Railroad to the Committee on Train Resistance and Tonnage Rating reporting to the 1916 convention of the American Railway Master Mechanics' Association.

METHODS OF ESTIMATING THE SPEED-PULL RELATION.

1. Dr. Goss' Method, applying only to engines using saturated steam. The formula:

$$T.E. = 161H/V = 619,000/V$$

since for the H8b, H is 3840 square feet.

TABLE 2, Data for Speed-pull Curve, Goss Method.

V (M.P.H.)	T.E. (pounds)
5	(123,800)*
10	(61,900)
15	41,300
20	30,950
25	24,700
30	20,630

2. Mr. Houston's Modification of the Goss Formula.

$$T. E. = 143 H/V = 550,000/V$$

TABLE 3, Data for Speed-pull Curve, Houston's Modification of the Goss Formula.

V (M.P.H.)	T.E. (Pounds)
5	(110,000)
10	(55,000)
15	36,660
20	27,500
25	22,000
30	18,330

* The use of the parentheses in Tables 2 and 3 indicates that the numbers thus enclosed are inapropriate for the locomotive under consideration, being higher than the adhesion limit. This limit is 52,750 pounds, assuming 25 percent adhesion.

3. The A. R. E. A. Method, for the saturated steam engine. The heating surface is 3840 square feet; if 4,000 pounds of coal per hour is burned, the coal per sq. ft. of heating surface per hour is 1.04 pounds. If this coal has a B.t.u. value of 14,000, the evaporation per pound of coal is 6.25 pounds of water. The hourly evaporation is 25,000 pounds, or 417 pounds per minute. For 24-inch cylinders, the steam per foot of stroke (at 205 pounds pressure) is 1.510 pounds, hence the steam required per revolution is 4x28/12x1.51= 14.07 pounds. Therefore M = 417/14.07 = 29.7 R.P.M., or for 62-inch drivers, 5.45 miles per hour. The horsepower at M is 25,000 / 38.30 or 652, and from this we have the tractive effort at M equal to 652 x 375 / 5.45 = 44,860 pounds. For other speeds:

TABLE 4, Data for Speed-pull Curve for

H8b Engine, A. R. E. A. Method.

Speed In "M's"	Speed In M.P.H.	Percent. of T. E. at M	Tractive Effort Pounds.
0	0	106.0%	47,500
.5 M	2.72	103.0	46,100
1.0 M	5.45	100.0	44,860
1.5 M	8.17	81.4	35,400
2.0 M	10.90	69.4	31,000
2.5 M	13.62	60.6	27,100
3.0 M	16.35	52.8	23,600
4.0 M	21.80	40.1	17,900
5.0 M	27.25	31.1	13,900
6.0 M	32.70	25.3	11,200

4. The A. R. E. A. Method, for the engine using superheated steam. The heating surface is 3030 square feet; the coal per square foot of heating surface per hour is 1.33 lb.; the evaporation per pound of coal, 6.10 pounds of water;

the hourly evaporation is 24,400 pounds, or 406 pounds per minute. The steam per foot of stroke is 0.956 pound, and per revolution, 4 x 28/12 x 0.956 = 8.85 pounds. Hence M = 406/8.85 = 46 R.P.M. = 8.5 M.P.H. The horsepower at M is 24,400/24.0 = 1016.6, and the corresponding tractive effort is 45,600 pounds. At other speeds:

TABLE 5, Data for Speed-pull Curve for H8sb Engine, A. R. E. A. Method.

SPEED In "M's"	In M.P.H.	Percent of T. E. at M	Tractive Effort, lb.
0	0	106.0%	48,400
1.0 M	8.5	100.0%	45,600
1.5 M	12.7	73.0	33,300
2.0 M	17.0	59.0	27,000
2.5 M	21.3	50.0	22,800
3.0 M	25.5	43.5	19.800
3.5 M	29.8	38.3	17,500

5. Mr. Williamson's Method, as applied to the saturated steam locomotive. The evaporation is

$$E = 0.68\ B\ H$$

where B is the B.t.u. value of the coal used in thousands, and H the fireside heating surface. For this engine, E = 0.68 x 14 x 3400 = 32,300 pounds per hour. The formula for tractive effort is

$$T.E. = 375E\ /VS = 12,112,500/VS$$

where S is the water rate or steam consumed per indicated horsepower hour. The values for S are found from the formula

$$S = 1500/R + 0.09\ R$$

where R is the speed in revolutions per minute.

TABLE 6, Data for Speed-pull Curve, Engine H8b, Williamson Method.

V M.P.H.	R R.P.M	.09R	1500/R	S	VS	Tractive Effort, lb.
5	27.1	2.4	55.4	57.8	289.0	40,500
10	54.2	4.9	27.6	32.5	325.0	37,300
15	81.3	7.3	18.4	25.7	385.5	31,400
20	108.4	9.8	13.9	23.7	474.0	25,600
25	135.5	12.2	11.1	23.3	582.5	20,800
30	162.6	14.6	9.1	23.7	711.5	17,050

6. Williamson's Method, applied to the superheater engine. In this case H = 2680, hence E = 0.68 x 2680 x 14 = 25,400 pounds of steam per hour. To determine the values for S, the steam consumption tables used in the A. R. E. A. method are used with the assumption that M = 10 miles per hour. The formula:

$$T. E. = 375 \times 25{,}400 / VS = 9{,}520{,}000/VS$$

TABLE 7, Data for Speed-pull Curve for Engine H8sb, Williamson Method.

V M.P.H.	S	VS	Tractive Effort, lb.
10	24.00	240.0	39,600
15	21.92	328.0	29,000
20	20.32	406.4	22,500
25	19.22	480.0	19,800
30	18.40	552.0	17,300

7. Mr. Houston's Method, saturated steam. The evaporation is 12.20 pounds of steam per square foot of heating surface per hour when 14,000 B.t.u. coal is used, if coal is fired at the rate of 1.75 pounds per square foot of heating surface per hour. To make this estimate

of evaporation it is assumed that the maximum firing rate which can be sustained is 5,000 lb. of coal per hour, or 1.30 lb. per sq. ft. of heating surface per hour. Reducing the assumed evaporation rate in direct ratio we have 12.2 x 1.30/1.75, or 9.05 pounds per square foot of heating surface per hour. The total evaporation is 3840 x 9.05 or 34,800 pounds, equivalent evaporation. Using the factor 1.207, the actual evaporation per hour is 28,800 pounds. The formula for tractive effort is

$$T.E. = 375\ E\ /\ S\ V = 10{,}800{,}000\ /\ S\ V$$

where S is the water-rate. The value of S is found from

$$S = 34.0 - .1R + .001\ R^2/4$$

where R is the speed in revolutions per minute.

TABLE 8, Data for Speed-pull Curve for Engine H8b, Houston's Method.

V M.P.H.	R R.P.M.	R/10	$R^2/4000$	S	SV	T. E. lb.
0	0	0	0	34.0	0	(∞)
5	27.1	2.7	0.2	31.5	157.5	(68,400)
10	54.2	5.4	0.7	29.3	293.0	36,900
15	81.3	8.1	1.7	27.6	415.0	26,000
20	108.4	10.8	2.9	26.1	522.0	20,600
25	135.5	13.6	4.6	25.0	625.0	17,250
30	162.6	16.3	6.6	24.3	729.0	14,800

8. Houston's Method, superheated steam. For the H8sb engine, the heating surface is 3030 square feet, and if 5,000 pounds per hour are fired, the combustion rate is 1.66 pounds per suare foot of heating surface per hour. The evaporation rate is then 12.20 x 1.66/1.75 or 11.55 pounds of steam (equivalent) per square foot of

heating surface per hour. Applying the evaporation factor 1.300, and multiplying by the heating surface the actual hourly evaporation is 27,000 pounds. The formula for tractive effort is then

$$T.E. = 375\ E / S\ V = 10{,}020{,}000 / S\ V$$

For the values of S, the figures given in the A.R.E.A. tables are used, assuming that M is 10 miles per hour.

TABLE 9, Data for Speed-pull Curve for Engine H8sb, Houston's Method.

V M.P.H.	S	VS	T.E. lb.
10.	24.00	240.0	42,500
15.	21.92	328.0	32,000
20.	20.32	406.4	25,200
25.	19.22	480.5	21,400
30.	18.40	552.0	18,500

10. The Kiesel Formula, saturated steam. Mr. Kiesel's general formula is as follows:

$$T.\ E. = \frac{2\ P\ M}{1 + \frac{110\ w}{3} \times \frac{M\ V}{K\ H}}$$

The symbols are as follows:

P = the initial pressure, ten pounds less than the working pressure of the boiler.

M = 1 d^2/ D, or the stroke of the cylinder times the diameter squared divided by the diameter of the drivers. (All dimensions in inches.)

w = the weight of one cubic foot of steam at initial pressure. (Pressure P.)

V = speed in miles per hour.

K = the evaporation rate per square foot of heating surface per hour. This rate varies from 10 to 15 pounds, the lower value applying to long runs.

H = the heating surface. The water-side of the tubes is used, and where a superheater is used, its surface is considered as part of the heating surface, and added accordingly.

For the H8b, we have the following values: P = 195 and M = 260, hence 2 P M = 101,440. Assuming a run of several hours in length, representing average freight-service conditions, K = 10. Since H= 3840, K H = 38,400. From these values, M V / K H = 0.006776 V . From steam tables, w = 0.456, therefore 110/3 x w = 16.72. Also 16.72 x 0.006776 V = 0.1133 V, and the formula becomes

$$T.\ E. = \frac{101,440}{1 + 0.1133\ V}$$

For this engine the tractive effort at the rim of the drivers (using the factor 85%) is 45,330 pounds. Allowing 5,000 pounds for machine friction the cylinder tractive effort at starting is 50,330 pounds. This value is used to finish the low-speed section of the speed-pull curve, where the calculated values are higher than the adhesion limit.

TABLE 10, Data for Speed-pull Curve, Engine H8b, Kiesel's Formula.

V M. P. H.	0.1133V	1 + 0.1133V	T. E. lb.
5	0.5665	1.566	(63,900)
10	1.1330	2.133	46,900
15	1.6995	2.700	37,000
20	2.2660	3.266	30,600
25	2.8300	3.830	26,100
30	3.3990	4.399	22,700

10. Kiesel's Formula, superheated steam. For the H8sb we have the following values: P = 195, M = 260, w = 0.328, H = 3030+780 = 3810, K = 10; therefore M V/ K H = 0.00683 V, 2 P M = 101,440, and 110w/3 = 11.9. Hence the formula becomes

$$T.E. = \frac{101,440}{1 + 0.0815\ V}$$

The same statement of the starting tractive effort applies as in the case of the saturated steam engine.

TABLE 11, Data for Speed-pull Curve for Engine H8sb, Kiesel's Formula.

V M. P. H.	0.0815 V	1 + 0.0815 V	T. E. lb.
5	0.4075	1.408	(72,000)
10	0.8150	1.815	(55,600)
15	1.2220	2.222	45,600
20	1.6300	2.630	38,600
25	2.0370	3.037	33,400
30	2.4450	3.445	29,400

11. Baldwin Method, saturated steam. For the H8b the rated tractive effort ($.85pld^2/D$) is 45,330 pounds. The heating surface is 3840 square feet, therefore T.E./H.S. is equal to 11.8. The curve for a ratio of 12 is used. The friction correction is based on the common estimate of 22 pounds per ton of weight on the drivers.

TABLE 12, Data for Speed-pull Curve, Engine H8b, Baldwin Method.

Speed M.P.H.	Fraction of Rated T.E.	T.E. at Rim, lb.	Friction lb.	Cyld.T.E. lb.
0	1.00	45,330	2250	47,580
5	.97	43,970	2250	46,420
10	.94	42,610	2250	44,860
15	.79	35,800	2250	38,050
20	.62	28,100	2250	30,350
25	.49	22,200	2250	24,450
30	.41	18,560	2250	20,810

12. Baldwin Method, superheated steam. For the H8sb the rated tractive effort is the same as for the H8b, and the heating surface is 3030 square feet, hence T.E./ H.S. is 9.3. To obtain the proper ordinates it is necessary to interpolate between the curves for ratios of 8 and 10. The same figures for friction are used.

TABLE 13, Data for Speed-pull Curve, Engine H8sb, Baldwin Method.

Speed M.P.H.	Fraction of Rated T.E.	T. E. at Rim, lb.	Friction lb.	Cyld.T.E. lb.
0	1.00	45,330	2250	47,580
5	.98	44,400	2250	46,650
10	.96	43,700	2250	45,950
15	.90	40,600	2250	42,850
20	.77	34,800	2250	37,050
25	.64	28,900	2250	31,150
30	.53	24,000	2250	26,250

13. The Friction Correction proposed by Mr. Henderson. Mr. Henderson's three methods of calculating the data for the speed-pull curve all give results in terms of the tractive effort at the rim of the drivers, and a friction correction must be used to obtain cylinder tractive effort. Mr. Henderson gives the following formula:

$$F = 0.15\ V + C$$

where F is the friction in percent. of the cylinder tractive effort, and C is a constant, an appropriate value for which in heavy freight service is 6.

TABLE 14, Henderson's Correction for Friction.

SPEED (V) M. P. H.	R. P. M.	0.15V	Percent. of Friction	Efficiency
11.05	60	1.66	7.66	92.3 %
13.30	75	2.07	8.07	91.9
16.60	90	2.50	8.50	91.5
22.10	120	3.31	9.31	90.7
27.60	150	4.15	10.15	89.8
33.20	180	5.00	11.00	89.0

The above values are calculated for certain speeds used in the three following tables.

14. Mr. Henderson's Three Speed-pull Curves. These curves apply only to locomotives using saturated steam. Table 15 gives the average curve presented in "Locomotive Operation " (p.377). Table 16 gives the data for the curve commonly known as the "Master Mechanics' Curve", presented by Mr. Henderson in a report to the 1898 convention of the Association. Table 17 gives the coordinates for a hyperbolic curve recommended in "Locomotive Operation". The formula for this curve is T R = 6000, where T is the

percentage of the theoretical tractive effort which can be be developed, and R is the speed in revolutions per minute.

TABLE 15, Data for Speed-pull Curve, Engine H8b, Henderson's Average Curve.

SPEED R. P. M.	SPEED M. P. H.	Percent. of T. T. E.	Tractive Effort, lb. At Rim	Tractive Effort, lb. In Cylinder
60	11.05	80	42,600	46,300
75	13.80	74	39,400	42,900
90	16.60	63	35,200	38,500
120	22.10	53	28,200	31,100
150	27.60	43	22,900	25,500
180	33.20	36	18,800	21,200

TABLE 16, Data for Speed-pull Curve, Engine H8b, Master Mechanics' Curve.

SPEED R. P. M.	SPEED M. P. H.	Percent. of T. T. E.	Tractive Effort, lb. At Rim	Tractive Effort, lb. In Cylinders
60	11.05	80	42,600	46,300
75	13.80	73	39,000	42,500
90	16.60	65	34,600	37,800
120	22.10	54	28,700	31,700
150	27.60	44	23,400	25,900
180	33.20	38	20,200	22,700

TABLE 17, Data for Speed-pull Curve, Engine H8b, Henderson's Hyperbola.

SPEED R. P. M.	SPEED M. P. H.	Percent. of T. T. E.	Tractive Effort, lb. At Rim	Tractive Effort, lb. In Cylinders
60	11.05	100	53,330	(57,800)
75	13.80	80	42,600	46,400
90	16.60	67	35,600	38,900
120	22.10	50	26,660	29,400
150	27.60	40	21,300	23,800
180	33.20	33	17,750	20,000

15. Data for Other Speed-factor Curves. The six following tables present the coordinates for the speed-pull curve of the H8b as calculated by five different methods, and one curve for the H8sb. For either engine, the theoretical tractive effort is 53,330 pounds, and the rated tractive effort (using a factor of 85%) is 45,330 pounds. Where a friction correction is required, the value used is 22 pounds per ton of weight on the drivers.

TABLE 18, Data for Speed-pull Curve, Engine H8b, Cole Speed-factor Method.

Piston Speed	Miles per Hour	Fraction of Rated T. E.	T. E. at Rim, lb.	Friction lb.	T. E. Cyld., lb.
0	0	1.00	45,330	2250	47,580
100	3.96	1.00	45,330	2250	47,580
200	7.92	1.00	45,330	2250	47,580
300	11.88	.95	43,060	2250	45,310
400	15.84	.86	38,980	2250	41,230
500	19.80	.77	35,130	2250	37,380
600	23.76	.68	30,820	2250	33,070
700	27.72	.59	26,740	2250	28,990
800	31.68	.52	23,510	2250	25,760

TABLE 19, Data for Speed-pull Curve, Engine H8sb, Cole Speed-factor Method.

Piston Speed	Miles per Hour	Fraction of Rated T. E.	T. E. at Rim, lb.	Friction lb.	T. E. Cyl., lb.
0-500	Values same as in Table 18.				
600	23.76	.68	30,820	2250	33,070
700	27.72	.61	27,600	2250	29,850
800	31.68	.54	24,400	2250	26,650

TABLE 20, Data for Speed-pull Curve, Engine H8b,

Curve Based on University of Illinois Tests.

Piston Speed	Miles per Hour	Fraction of T. T. E.	T. E. at Rim, lb.	Friction lb.	T. E. in Cylds., lb.
100	3.96	0.88	46,930	2250	49,180
200	7.92	.85	45,330	2250	47,580
300	11.88	.81	43,200	2250	45,450
400	15.84	.73	38,930	2250	41,180
500	19.82	.64	34,130	2250	36,380
600	23.76	.55	29,330	2250	31,580
700	27.72	.47	25,060	2250	27,310
800	31.68	.41	21,860	2250	24,110

TABLE 21, Data for Speed-pull Curve, Engine H8b,

Baldwin Speed-factors.

Piston Speed	Miles per Hour	Fraction of T. T. E.	Cylinder T. E. lb.
0	0	0.97	51,600
100	3.96	.96	51,000
200	7.92	.87	46,200
300	11.88	.76	40,500
400	15.84	.65	34,600
500	19.80	.54	28,800
600	23.76	.42	22,300
700	27.72	.31	16,500
800	31.68	.25	13,300

TABLE 22, Data for Speed-pull Curve, Engine H8b,

Schenectady Loco. Co. Speed-factors.

Piston Speed	Miles per Hour	Fraction of T. T. E.	Cylinder T.E. lb.
0	0	0.85	45,330
100	3.96	.85	45,330
200	7.92	.85	45,330
300	11.88	.80	42,500
400	15.84	.71	37,800
500	19.80	.62	33,000
600	23.76	.54	28,600
700	27.72	.47	25,000
800	31.68	.41	21,800

TABLE 23, Data for Speed-pull Curve, Engine H8b, Adams' & Isaacs' Method.

Piston Speed	Miles per Hour	Fraction of T.T.E.	Cylinder T. E., lb.
0	0	0.950	50,500
500	19.80	.632	33,600

Two points only are required to define the above relation, since it is a straight line.

APPENDIX III-A

A STUDY OF THE A. R. E. A. METHOD UNDER VARIOUS OPERATING CONDITIONS.

The curves of Fig. 7 are all based upon an engine having the following general dimensions:

Cylinders,	24 x 30"
Diameter of Drivers,	60"
Working Pressure,	200#
Theoretical T. E.,	57,600#
Total heating surface,	3244 sq.ft.
Superheating surface,	700 sq.ft.
Tubes, No., Daimeter,	280--2"
Flues, No., Diameter,	36--5 3/8"
Length, Tubes and Flues,	15'6"
Grate Area,	65 sq. ft.

The normal rate of firing is 4000 pounds of coal per hour, the coal having a heating value of 13,000 B.t.u. The normal dimensions and conditions are varied in eight different ways, each of the special conditions or dimensions superseding

Table 24

Effect of Varying Conditions as Shown by the A.R.E.A. Method

Item	Case 1 Normal	Case 2 H.S. 4400	Case 3 Cyld. 26×30	Case 4 3000# Coal	Case 5 5000# Coal	Case 6 15000 Btu	Case 7 Leak	Case 8 E = 35,000	Case 9 Saturated
1. Coal per Hour, lb.	4000	4000	4000	3000	5000	4000	4000		4000
2. Quality, B.t.u.	13000	13000	13000	13000	13000	15000	13000		13000
3 Heating Surface, sq. ft.	3240	4400	3240	3240	3240	3240	Normal Evaporation, — 4000 =		3800
4. Coal per Hour per sq. ft. H.S., lb.,	1.23	.90	1.23	.92	1.54	1.23			1.05
5. Evaporation per lb. Coal, lb.	5.85	6.57	5.85	6.53	5.28	6.75			6.23
6. Total Evap. per Hour, lb.,	23,400	24,500	23,400	19,800	26,400	27,000	19,400	35,000	24,920
7. Evap. per Minute, lb.,	390	438	390	330	440	450	323	583	415
8. Steam per foot of stroke, lb.,	.931	.931	1.097	.931	.931	.931	.931	.931	1.477
9. Steam per Revolution, lb.,	9.31	9.31	10.97	9.31	9.31	9.31	9.31	9.31	14.77
10. R.P.M. for "M"	41.9	47.2	35.5	35.5	47.2	48.0	34.5	62.5	28.0
11. "M" in M.P.H.	7.3	8.45	6.35	6.35	8.40	8.60	6.15	11.25	5.00
12. Horsepower at M	975	1095	975	8.25	1100	1125	812	14,600	650
13. Tractive Effort at M	50,300	48,500	57,100	48,600	49,100	49,100	49,500	48,500	48,700
14 Miles per Hour	0	0	0	0	0	0	0	0	0
Fraction of T.E. at M .0M	1.06	1.06	1.06	1.06	1.06	1.06	1.06	1.06	1.06
Tractive Effort	53,300	51,500	60,500	51,600	52,100	52,100	52,500	51,500	51,700
15 Ditto .5M	3.65	4.22	3.17	3.17	4.20	4.30	3.07	5.62	2.50
	1.03	1.03	1.03	1.03	1.03	1.03	1.03	1.03	1.03
	51,900	50,000	59,000	50,000	50,600	50,600	51,000	50,000	50,200
16 Ditto 1.M	7.3	8.45	6.35	6.35	8.40	8.60	6.15	11.25	5.00
	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00
	50,300	48,500	57,100	48,600	49,100	49,100	49,500	48,500	48,700
17. Ditto 1.2M	8.76	9.86	7.65	7.65	10.20	10.30	7.30	13.50	6.00
	.865	.865	.865	.865	.865	.865	.865	.865	.915
	43,300	42,000	49,200	42,000	42,500	42,500	43,000	42,000	44,600
18. Ditto 1.4M	10.21	11.82	8.90	8.90	11.80	12.00	8.60	15.75	7.00
	.769	.769	.769	.769	.769	.769	.769	.769	.845
	38,500	37,300	44,000	37,300	37,900	37,900	38,500	37,300	41,200
19. Ditto 1.6M	11.69	13.50	10.19	10.19	13.40	13.70	9.85	18.00	8.00
	.695	.695	.696	.696	.696	.696	.696	.696	.786
	34,800	33,800	39,800	33,800	34,200	34,200	34,800	33,800	38,400
20. Ditto 2.M	14.60	16.90	12.70	12.70	16.80	17.20	12.30	22.50	10.00
	.590	.590	.590	.590	.590	.590	.590	.590	.694
	29,500	28,600	33,900	28,600	29,000	29,000	29,500	28,600	33,700
21. Ditto 2.5M	18.25	21.10	15.90	15.40	21.0	21.50	15.38	28.12	12.50
	.500	.500	.500	.500	.500	.500	.500	.500	.606
	25,150	24,200	28,500	24,300	24,550	24,550	24,750	24,250	29,500
22. Ditto 3.M	21.90	25.30	19.05	19.05	25.20	25.80	18.50	33.75	15.00
	.435	.435	.435	.435	.435	.435	.435	.435	.528
	21,750	21,100	24,900	21,100	21,400	21,400	21,700	21,100	25,700
23. Ditto 4.M	29.20	33.80	25.40	25.40	33.60	34.40	24.60	45.00	20.00
	.344	.344	.344	.344	.344	.344	.344	.344	.401
	17,410	16,600	19,700	16,600	16,900	16,900	17,200	16,600	19,500
24. Ditto 6.M	43.80	50.60	37.60	37.60	50.40	51.60	36.90	67.50	30.00
	.240	.240	.240	.240	.240	.240	.240	.240	.253
	12,000	11,700	13,700	11,700	11,800	11,800	12,000	11,700	12,300

some condition or combination of conditions of the normal case. In every case, all factors remain as in the original except those specified. The nine curves plotted represent the following conditions, Case 1 being the normal curve:

Case 2, Heating surface increased to 4400 sq. ft.

Case 3, Cylinder diameter increased to 26 inches.

Case 4, Firing Rate, 3000 pounds of coal per hour.

Case 5, Firing Rate, 5000 pounds of coal per hour.

Case 6, Coal of 15,000 B.t.u. used.

Case 7, Loss of 4000 lb. steam per hour in leakage or otherwise.

Case 8, Increase of evaporation estimate to 35,000 lb. per hour.

Case 9, Engine without superheater, the total heating surface being 3800 sq. ft.

All of the steps in calculating the data for the various curves are given in Table 24. The methods of calculation are further exemplified in Appendix II - A.

APPENDIX III-B

EVAPORATION ESTIMATES.

The boiler of the locomotive of the foregoing section has the following dimensions:

Heating surface	3240 sq. ft.
Superheating surface	700 sq. ft.
Grate Area	65 sq. ft.
Tube length	15.5 feet.

There are 270 tubes of 2-inch diameter, and 36 flues 5 3/8 inches in diameter.

By Mr. Cole's estimate, the evaporation for this boiler is as follows: If tube-spacing of $\frac{3}{4}$ inch is assumed, the evaporation per square foot of area of a flue of this length and diameter is 9.80 lb. steam per hour. The tube heating surface is 2200 sq. ft. For the flues, 800 square feet of surface, and an evaporation rate of 11.41 lb. For the firebox, with 240 square feet of heating surface, an evaporation rate of 55 lb. per square foot per hour. Summary:

Tubes, 2200 x 9.80lb.	=	21,600 lb.
Flues, 800 x 11.45 lb.	=	9,150 "
F'box, 240 x 55.00 lb.	=	13,200 "
Total		43,950 lb.

The firing rate upon which this estimate is based is indefinite, but is presumably between 100 and 125 lb. per sq.ft. of grate per hour. The quality of coal is not taken into account: if this estimate is reduced by the coal factor applicable (for 13,000 B.t.u. coal, the factor is 0.87, as shown in Section VII) it becomes 38,100 lb. per hour.

Mr. Williamson's estimate:

$$0.68 \times 13 \times 3240 = 28,600 \text{ lb. per hour}$$

if a firing rate of 1 lb. coal per hour per square foot of heating surface is assumed. If it is assumed that 4000 lb.

of coal per hour is to be fired, the estimate is

$$0.68 \times 13 \times 4000 = 35{,}500 \text{ lb. per hour.}$$

The use of this method has been explained in Section II and in Appendix II-A.

Mr. Houston's estimate: Coal may be fired at the rate of 1.55 lb. per sq. ft. of heating surface per hour, therefore the evaporation rate for 13,000 B.t.u. coal must be reduced. The rate applying to 13,000 B.t.u. coal and a firing rate of 1.75 pounds per sq. ft. heating surface per hour is 11.5 pounds of steam per sq. ft. of heating surface. For the lower firing rate, $11.5 \times 1.55/1.75 = 10.15$ lb. Hence the total actual evaporation is

$$10.15 \times 3240 \times 1/1.2 = 28{,}800 \text{ lb. per hour.}$$

The estimate by the A. R. E. A. method is 23,400 lb. per hour, as previously worked out. For the eighth special condition of Appendix III-A, an evaporation of 35,000 lb. per hour has been assumed: this is 10.8 lb. per square foot of heating surface per hour, apparently not an excessive rate.

APPENDIX V-A

TABLE 25, Dimensions of Pennsylvania Superheater Locomotives, Tested in the Company's Laboratory at Altoona.

Class and Type	Cylinders	Working Pr.
E3sd, Atlantic	22 x 26"	205 lb.
E6s,(old), Atlantic,	22 x 26"	205 lb.
E6s,(new), Atlantic,	$23\frac{1}{2}$ x 26"	205 lb.
K29s, Pacific,	27 x 28"	200 lb.
K2as, Pacific,	24 x 26"	205 lb.
K4s, Pacific,	27 x 28"	205 lb.
H8sb, Consolidation,*	25 x 28"	205 lb.
L1s, Mikado,	27 x 30"	205 lb.

Class	Diameter of Drivers	Theoretical T. E. (T_o)	Evaporation E
E3sd	80"	32,200 lb.	42,000 lb.
E6s (old)	80"	32,200 lb.	51,000 lb.
E6s (new)	80"	35,600 lb.	55,000 lb.
K29s	80"	51,000 lb.	67,000 lb.
K2as	80"	38,400 lb.	58,000 lb.
K4s	80"	52,400 lb.	84,000 lb.
H8sb	62"	57,600 lb.	43,000 lb.
L1s	62"	72,100 lb.	76,000 lb.

All necessary information concerning the heating surface of these engines appears in Table 50.

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* The engine tested was not strictly of the class named, as the H8sb has cylinders 24 inches in diameter. In Bulletin 10 of the Pennsylvania Test Department, the engine is referred to as a "modified H8sb", the modification consisting of the increased cylinder diameter.

APPENDIX V-B

TABLE 26, Data for Speed-pull Curves from Laboratory Tests, Superheater Locomotives.

M.P.H.	Piston Speed	Drawbar Pull, lb.	Friction lb.	Cyld. T.E. lb.	Fraction of T_o	Fraction of E
Engine H8sb, Consolidation, T_o/E = 1.340, T_o=57,600, E= 43,000.						
7.2	187	49,900	4800	54,700	0.950	1.275
10.8	280	44,300	4800	49,100	.850	1.150
14.4	374	37,500	4700	41,200	.716	.960
18.0	466	33,100	4700	37,800	.655	.880
21.6	560	27,200	4600	31,800	.550	.740
25.2	664	22,800	4600	27,400	.475	.635
28.7	748	19,900	4500	24,400	.406	.565
30.5	795	17,800	4400	22,200	.386	.518
45.0	1130			(16,500)		(.409)
Engine K29, Pacific, T_o/E=0.760, T_o=51,000, E=67,000.						
37.9	746	21,250	2300	23,550	0.464	0.352
42.7	840	19,100	2300	21,400	.420	.320
47.3	933	17,350	2500	19,850	.370	.295
56.9	1120	14,100	2800	16,900	.320	.251
66.3	1310	11,600	3300	14,900	.290	.222
Engine E3sd, Atlantic, T_o/E=0.770, T_o=32,200, E=42,000						
37.3	695	16,300	2100	18,400	0.570	0.439
46.7	866	13,300	2100	15,400	.476	.366
56.0	1040	10,700	2400	13,100	.406	.312
74.7	1390	(8,000)	3200	(11,200)	(.346)	(.267)
Eng. E6s(new), Atlantic, T_o/E=0.655, T_o=36,000, E=55,000.						
14.0	260	30,900	2200	33,100	0.930	0.600
18.7	366	30,000	2000	32,000	.960	.580
23.4	433	28,700	2000	30,700	.861	.560
28.0	520	25,300	1800	27,100	.760	.494
37.4	733	20,700	1500	22,200	.630	.404
46.8	866	17,400	1700	19,100	.535	.345
56.0	1040	14,800	1800	16,600	.466	.300
65.8	1210	12,200	1900	14,100	.397	.256
74.8	1390	10,200	2300	12,500	.351	.226

Table 26, Continued.

M.P.H.	Piston Speed	Drawbar Pull, lb.	Friction lb.	Cyld. T.E. lb.	Fraction of T_o	Fraction of E
E6s, (old), Atlantic, $T_o/E=0.633$, $T_o=32,200$, $E=51,000$.						
37.7	694	17,400	2300	19,700	0.610	0.386
47.1	866	14,800	2300	17,100	.530	.355
56.5	1040	12.850	2700	15,550	.480	.304
65.7	1216	11,150	2700	13,850	.430	.271
75.0	1390	9,400	2800	12,200	.376	.240
K2as, Pacific, $T_o/E=0.660$, $T_o=38,400$, $E=58,000$.						
37.3	685	19,000	2800	21,800	0.558	0.378
46.5	866	16,840	1900	18,700	.486	.324
55.8	1041	14,000	2000	16,000	.416	.275
65.1	1210	11,200	2200	13,400	.360	.236
74.4	1390	(10,600)	2700	(13,300)	(.346)	(.228)
K4s, Pacific, $T_o/E=0.625$, $T_o=52,400$, $E=84,000$.						
30.0	588	34,000	2500	36,500	0.691	0.435
40.0	784	27,400	2700	30,100	.572	.360
50.0	980	22,400	2900	25,300	.481	.302
60.0	1176	18,400	3700	21,600	.411	.267
70.0	1372	14,600	3500	18,400	.345	.215
L1s, Mikado, $T_o/E=0.945$, $T_o=72,000$, $E=76,000$.						
6.0	163	59,000	7000	66,000	0.920	0.870
10.0	271	57,000	4100	61,100	.850	.805
15.0	406	52,300	2700	55,000	.765	.725
20.0	542	45,500	2500	48,000	.670	.630
25.0	678	38,000	2700	40,700	.565	.535
30.0	814	32,400	3000	35,400	.490	.465
45.0	1221			(24,200)	(.335)	(.314)

APPENDIX V-C

TABLE 27, Data for Indicated Horsepower Curve,

Engine H8sb.

Cylinder T. E.,lb.	M. P. H.	Piston Speed	Indicated Horsepower
54,700	7.2	187	1055
49,100	10.8	280	1410
41,200	14.4	374	1570
37,800	18.0	466	1820
31,800	21.6	560	1830
27,400	25.2	664	1840
24,400	28.7	748	1875
22,200	30.5	795	1930
Extrapolated:			
16,500	45.0	1130	1980

TABLE 28, Data for Indicated Horsepower Curve,

Engine L1s.

Cylinder T. E.,lb.	M. P. H.	Piston Speed	Indicated Horsepower
66,000	6.0	163	1060
61,000	10.0	271	1630
55,000	15.0	406	2400
48,000	20.0	542	2560
40,700	25.0	678	2610
35,400	30.0	814	2850
Extrapolated:			
23,900	45.0	1220	2880

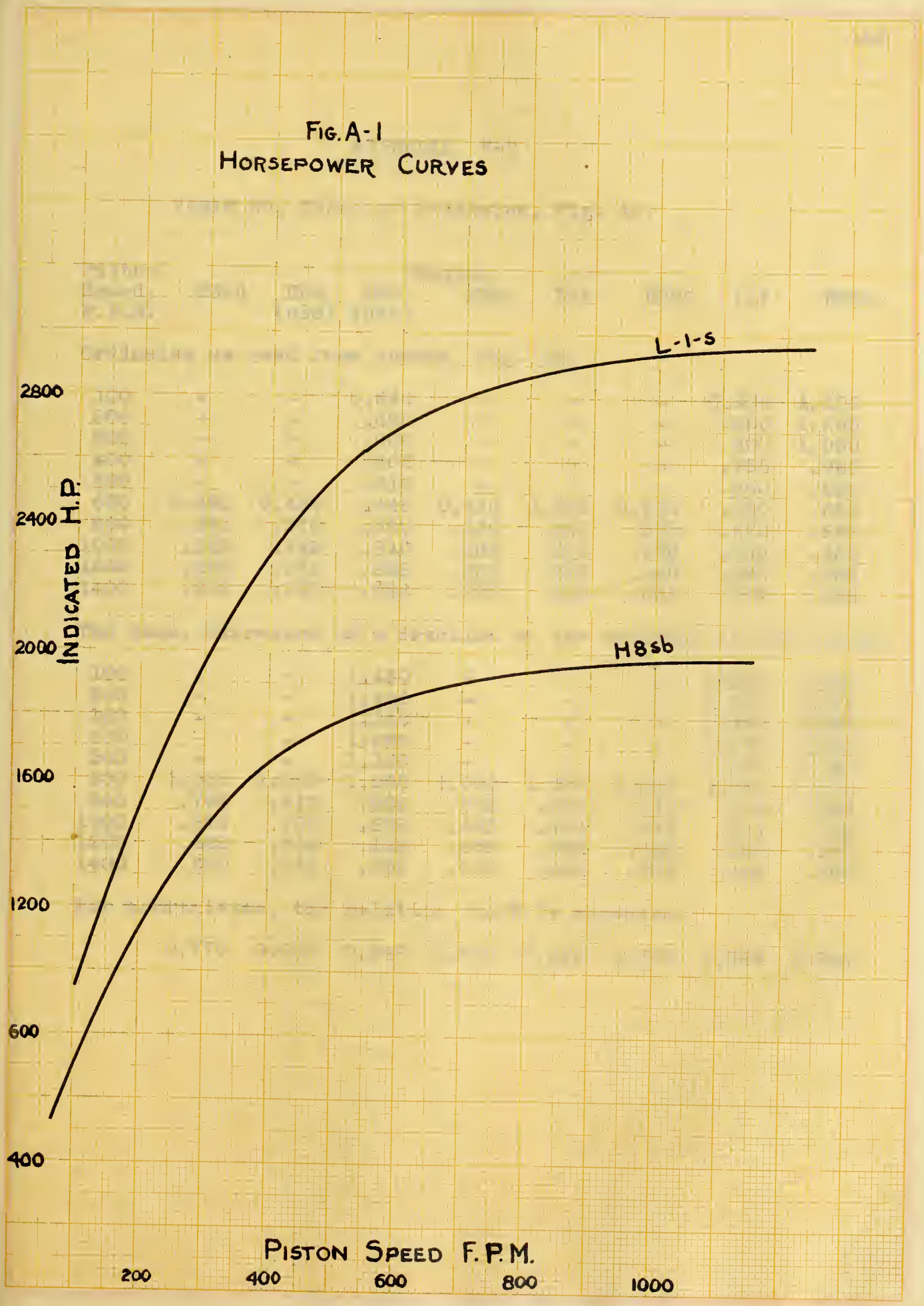
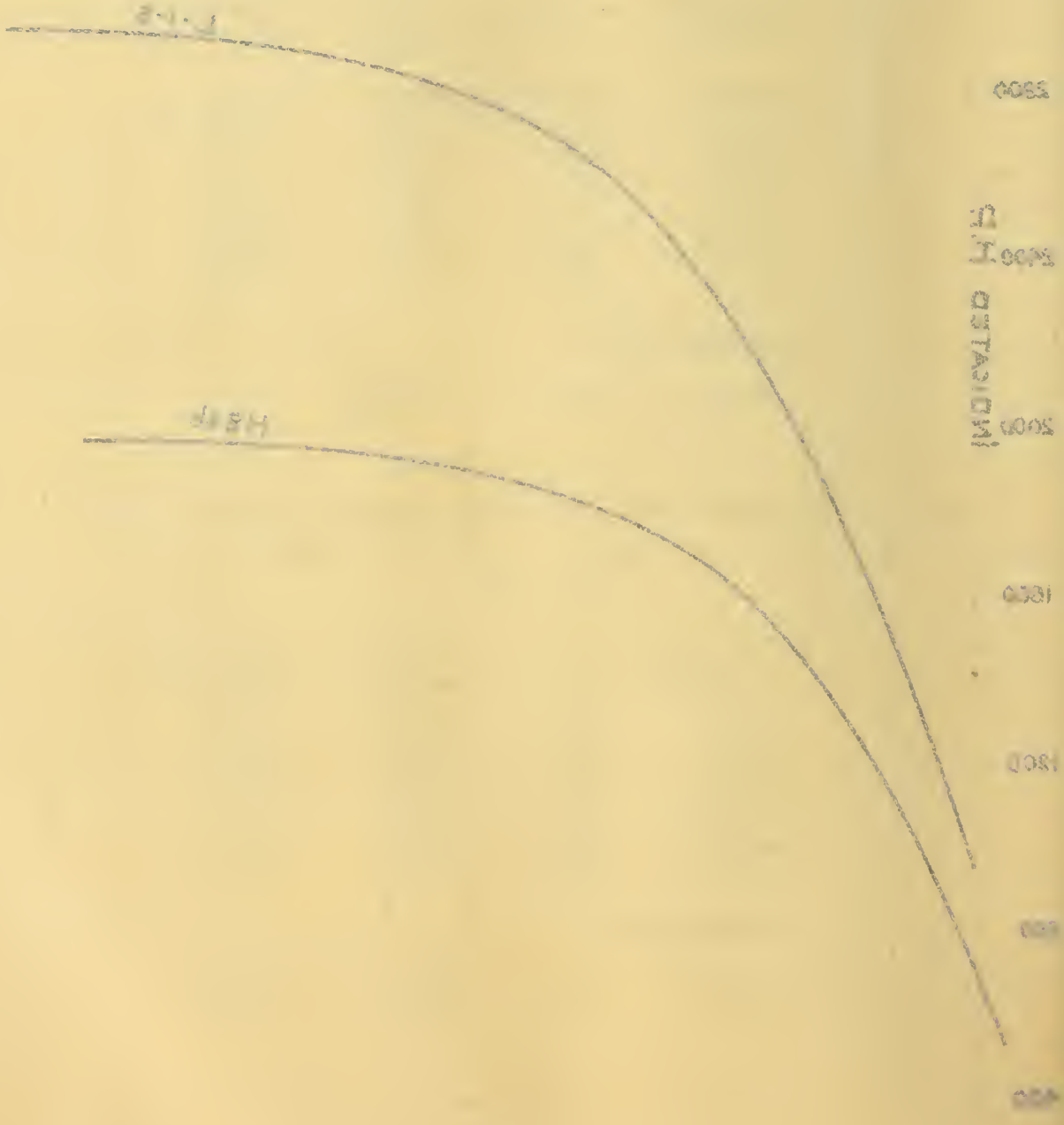


FIG. A-1
HORSEPOWER CURVES
L-1-S
H8sb
INDICATED H.P.
2800
2400
2000
1600
1200
600
400
PISTON SPEED F.P.M.
200
400
600
800
1000



APPENDIX V-D

TABLE 29, Table of Ordinates, Fig. 12.

Piston Speed, F.P.M.	Engine E3sd	E6s (old)	E6s (new)	K2as	K4s	K29s	L1s	H8sb
Ordinates as read from curves, Fig. 12:								
100	-	-	0.640	-	-	-	0.904	1.306
200	-	-	.625	-	-	-	.860	1.255
300	-	-	.600	-	-	-	.800	1.000
400	-	-	.565	-	-	-	.735	.940
500	-	-	.510	-	-	-	.660	.800
600	0.490	0.450	.445	0.410	0.430	0.410	.580	.690
800	.391	.370	.360	.330	.352	.332	.470	.542
1000	.330	.320	.310	.285	.300	.278	.385	.455
1200	.285	.272	.262	.250	.250	.240	.325	.385
1400	.255	.240	.226	.222	.210	.215	.274	.335
The same, expressed as a fraction of the ordinate at 600 f.p.m.:								
100	-	-	1.430	-	-	-	1.570	1.890
200	-	-	1.400	-	-	-	1.490	1.820
300	-	-	1.340	-	-	-	1.380	1.480
400	-	-	1.260	-	-	-	1.270	1.360
500	-	-	1.150	-	-	-	1.140	1.160
600	1.000	1.000	1.000	1.000	1.000	1.000	1.000	1.000
800	.798	.819	.805	.802	.824	.810	.810	.786
1000	.674	.720	.692	.695	.701	.679	.664	.662
1200	.582	.604	.588	.609	.585	.586	.560	.560
1400	.520	.530	.505	.540	.494	.524	.466	.485
For convenience, the relation T_o/E is repeated:								
	0.770	0.633	0.649	0.660	0.625	0.760	0.948	1.340

APPENDIX V-E

TABLE 30, Relation Between Tractive Effort at 600 Feet per Minute Piston Speed, and the Ratio T_o / E .

Engine	T_o	E	T_o/E	t/E	t/T_o
E3sd	32,200	42,000	0.770	0.490	0.635
E6s (old)	32,200	51,000	.633	.450	.705
E6s (new)	36,000	55,000	.655	.445	.682
K2as	38,400	58,000	.660	.410	.620
K4s	52,400	84,000	.625	.430	.690
K29s	51,000	67,000	.760	.410	.540
L1s	72,000	76,000	.948	.580	.610
H8sb	57,600	43,000	1.340	.690	.515

For convenience the interpretation of the symbols is repeated:

T_o = the theoretical tractive effort, or pld^2/D

E = the normal maximum hourly rate of evaporation (equivalent), depending on the design of the boiler, the rate of firing, and the quality of the fuel.

t = the cylinder tractive effort in pounds which can be developed at a piston speed of 600 feet per minute.

APPENDIX V-F

CHECKING POINTS FOR FIG. 13.

In order to extend the curve of Fig. 13 and to verify its form it is necessary to find test data for engines with a higher value of the T_o/E ratio than the H8sb (1.340) and a lower value than the K4s and the E6s(old), the ratios for these engines being 0.625 and 0.633 respectively. The former should be possible, as the use of poor coal or otherwise limited boiler capacity might reduce the evaporation of an engine with a larger T_o considerably below that of the H8sb. The second case mentioned may be assumed to be impossible, for in the cases of the two engines named, the K4s had an extremely large evaporation, produced by a rate of firing hardly possible in road service, and the E6s has cylinders which are very small in comparison with the boiler capacity.

A method of extending the data in the second case is suggested by the curves for tractive effort at short cut-offs which are plotted in the bulletins of the Pennsylvania Test Department. If these curves be extended to zero-speed they may be considered as representing the performance of an engine having the same boiler capacity as the engine from which they were taken, but smaller cylinders--that is, smaller T_o. For example, assume an engine having T_o equal to 28,000 lb.; this engine developes a pull of 20,000 lb. at 600 f.p.m. piston speed, the cut-off being 50%. At the same speed, the pull

is 15,000 lb., when the cut-off is 25%. Now if we assume that 15,000 lb. is the maximum pull which can be produced at this speed, the value of T_o must be smaller than in the true case. Assuming that the relation between the theoretical tractive effort and the tractive effort at 600 f.p.m. is constant, the value of T_o for the new condition would be x in the proportion 28,000:20,000:: x :15,000, or 21,000 lb. We know, however that the assumption made is not a proper one: for a large boiler capacity (i.e., a large E relative to T_o, or a small T_o/E ratio) the value of the ratio t/T_o is greater. If the curve of Fig. 13 is extended mechanically, simply by lengthening it in the manner suggested by the geometric form of the known portion, a more nearly accurate estimate of the existing relationship is obtained: by applying this estimate to an actual case, the correctness or incorrectness of the curve as extended will be shown. This process is carried out as follows:

On p. 134 of Bulletin 21 are plotted the drawbar-pull curves of the older E6s. For this engine T_o= 32,200 lb. At 600 f.p.m. piston speed, corresponding to 33 miles per hour, the maximum pull is 24,000, the cut-off being 50%. For this speed and cut-off the machine friction is 1700 lb., hence the cylinder tractive effort is 25,700 lb. At a cut-off of 25% the drawbar-pull is 10,400 lb., the machine friction 2200 lb., and the cylinder tractive effort 12,600 lb. Then, if the T_o rating were such that at this speed the maximum tractive effort were 12,600 lb., the value of T_o would be (by direct

proportion) 32,220 x 12,600/25,700 = 15,800 lb. Since E is 51,000 lb., the new value of T_o/E is 0.31. From the curve (Fig. 13) extended, when T_o/E is 0.31, t/T_o= 0.775. This gives a check on the assumption of the value of T_o, for the value of T_o is evidently 12,600/0.775 = 16,300 lb., showing the direct-proportion assumption to have been 500 lb. in error. Now check the values against the curve again: we have T_o/E = 16,300/51,000 = 0.32, and we already have t/T_o = 0.775 or 77.5%. This point lies exactly on the curve as extended.

It is well to show that the preceding process is not the mere repetition of the same figures in different terms, or "working in a circle". For instance, let it be assumed that by some process it has been determined that the value of T_o for the short cut-off condition is 15,000 lb. Now we have T_o/E = 0.294, and t/T_o = 0.84. This point is materially out of agreement with the extended curve, and with any other extension which can be justified by the form of the known portion of the curve.

In the same manner another checking-point is worked out for the E6s, the cylinder tractive effort being 9800 lb.,the cut-off 20%, and the speed 600 f.p.m. as before. Let it be assumed that the ratio T_o/E will be 0.24; the corresponding value of t/T_o from the extended curve is 0.8; from this we have T_o = 9800/0.8 = 12,200 lb. The T_o/E = 12,200/51,000 or 0.24, checking exactly with the assumption, and showing that the extension of the curve is approximately correct.

In the opposite case, that is,where the T_o/E ratio is

large, test data* furnish an opportunity to establish another checking-point. The engine in question developed 1800 indicated horsepower, and the evaporation corresponding, based on a water-rate of 26.6 lb. (equivalent) is 48,000 lb. The water-rate estimate is based upon Mr. Cole's figure of 20.8 lb.,(actual steam), and upon the Pennsylvania Mikado L1s, of similar size and type. The most representative water-rate for this engine (Bulletin 28) is 20 lb., actual steam. For the engine previously referred to, T_o= 70,000 lb., t = 34,400 lb., and as stated, E = 48,000 lb. Hence t/T_o is 0.493, and T_o/E = 1.460. This point furnishes a check on the extension of the curve of Fig. 13 to the right.

---

* The figures used are taken from road-tests of the A.T.& S.F. Mikado type engine 3100, having 25 x 32-inch cylinders, 57-inch drivers, and a working pressure of 200 lb. The data are available to the writer through the kindness of Mr. F.J. Prout of the Test Department staff, with the approval of Mr. H. B. Macfarland, Engineer of Tests.

APPENDIX V-G

SPECIMEN CALCULATIONS FOR SPEED-PULL CURVE

Given a locomotive having the following dimensions and operating conditions:

Cylinders	25 x 32"
Diameter of drivers,	60"
Working pressure	180 lb.
$T_o(pld^2/D)$	60,000 lb.
Evaporation per hour	63,000 lb.

From the given figures we have $T_o/E = 0.95$, hence from Fig. 13 $t/T_o = 0.6$, and $t = 0.6 \times 60{,}000 = 36{,}000$ lb., the cylinder tractive effort at 600 f.p.m. piston speed. Referring to Fig. 14, and following the abcissa 0.95, the relation of the tractive efforts at other speeds is found, that is, the ratio of the tractive effort at any other speed to that at 600 f.p.m.

TABLE 31, Data for Speed-pull Curve,

Proposed Method.

Piston Speed	Miles per Hour	Ordinate	Cylinder T. E.,lb.
100	3.35	1.55	56,000
200	6.70	1.49	53,640
300	10.05	1.37	49,400
400	13.40	1.27	45,690
500	16.75	1.15	41,500
600	20.10	1.00	36,000
800	26.80	.80	28,800
1000	33.50	.68	24,500
1200	40.20	.57	20,520
1400	46.90	.49	17,640

This curve is plotted in Fig. A-2.

APPENDIX V-H

APPLICATION OF THE PROCESS TO THE DATA USED IN ITS CONSTRUCTION.

The poorest agreement with the curves of Figs. 13 and 14 is shown in the points representing the engine K29s. This engine showed a low mean effective pressure at all points on its known curve, due to pressure reduction through the superheater. The disparity (averaging 13%) between the actual and the calculated results shows the effect of an abnormality of design: superheater tubes of excessive length. In Fig. A-2 the actual and estimated curves are compared. The data for the latter are as follows:

T_o= 51,000; E= 67,000; T_o/E = 0.760, hence t/T_o=0.63, and t = 32,000 lb.

TABLE 32, Data for Calculated Speed-pull Curve

Engine K29s.

Piston Speed	Miles per Hour	Ordinate	Cylinder T. E.,lb.
100	5.1	1.48	47,400
200	10.2	1.43	46,000
300	15.3	1.35	43,200
400	20.4	1.26	40,300
500	25.5	1.14	36,500
600	30.6	1.00	32,000
800	40.8	.81	26,000
1000	51.0	.69	22,000
1200	61.2	.59	19,000
1400	71.4	.51	16,300

In order to show the general agreement of the data

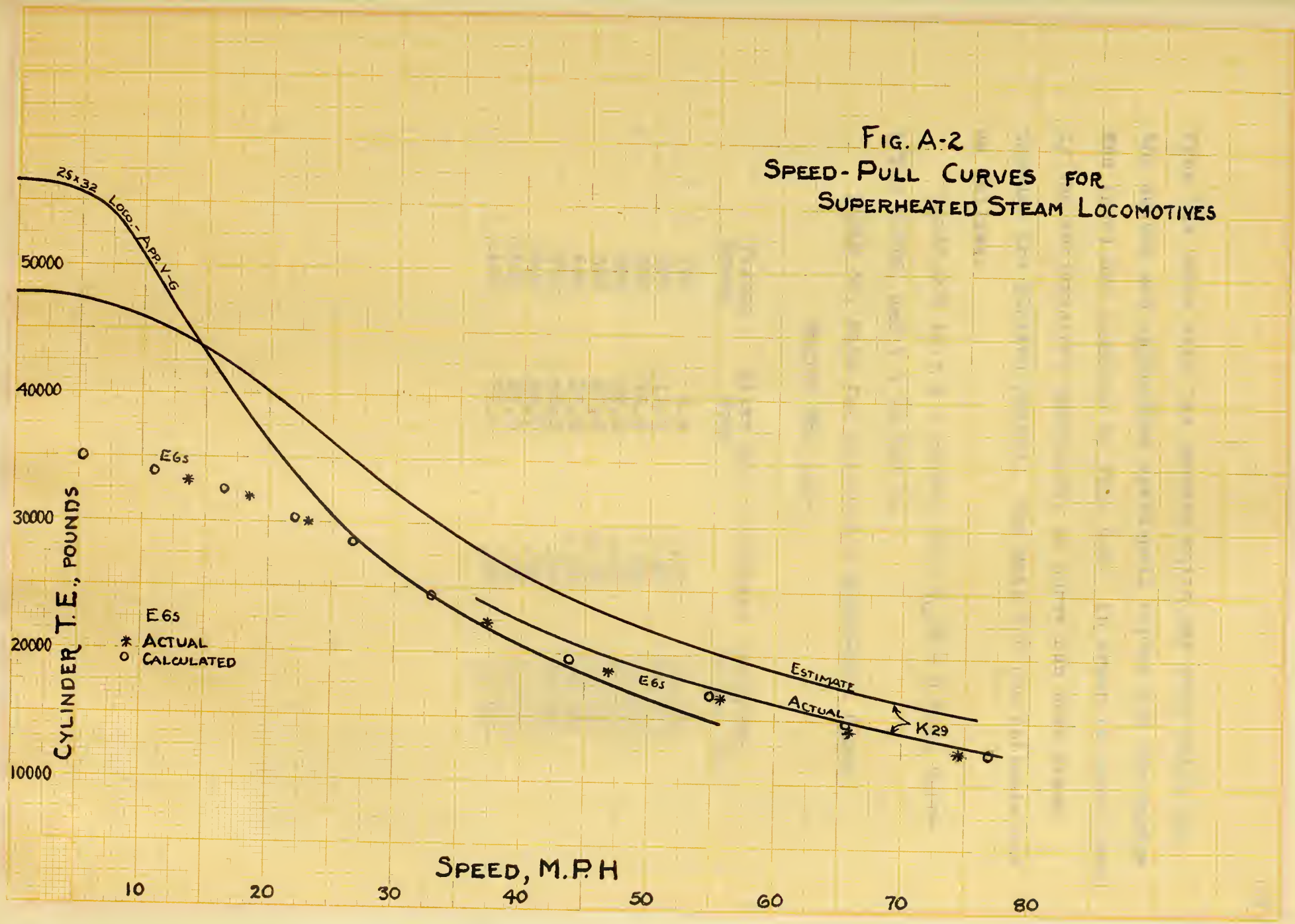
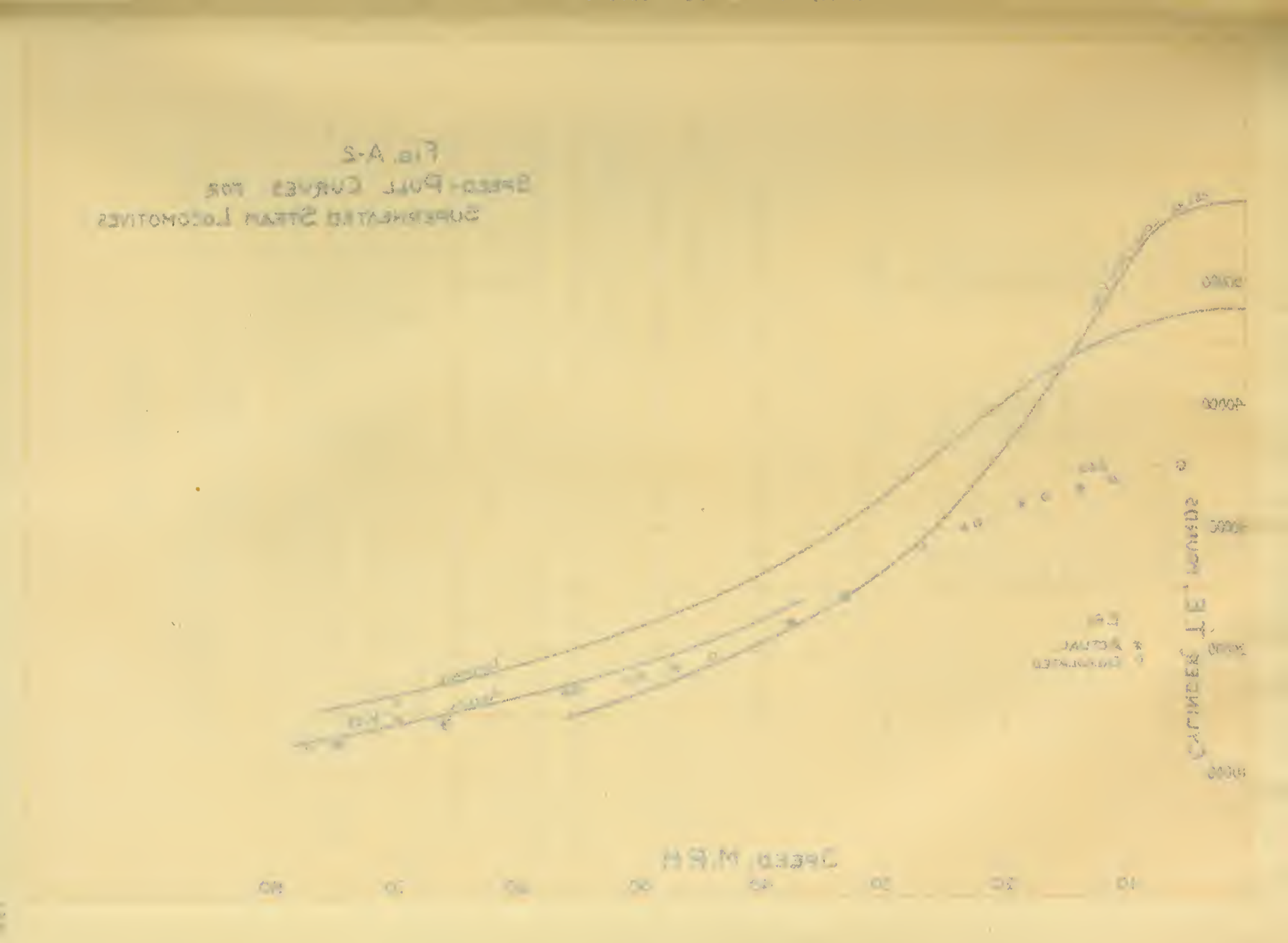


Fig. A-2
Speed-Pull Curves for
Superheated Steam Locomotives



from the tests with the process which has been built up, the actual and calculated speed-pull curves for the engine E6s (new) are compared in Fig. A-2. In order to show clearly the satisfactory agreement, no curve has been drawn through the plotted points. The data for the estimate are as follows:

T_o=36,000 lb.; E = 55,000 lb.; T_o/E = 0.655, hence t/T_o= 0.675, and t = 24,300 lb.

TABLE 33, Data for Calculated Speed-pull Curve, Engine E6s (new).

Piston Speed	Miles per Hour	Ordinate	Cylinder T. E., lb.
100	5.5	1.44	35,000
200	11.0	1.40	34,000
300	16.5	1.34	32,600
400	22.0	1.25	30,400
500	27.5	1.14	28,700
600	33.0	1.00	24,300
800	44.0	.82	19,900
1000	55.0	.71	17,200
1200	66.0	.59	14,300
1400	77.0	.52	12,600

APPENDIX VI-A

TABLE 34, Saturated Steam Locomotives for Which Laboratory Data are Available.

Designation, Road, and Type	Steam Distribution Class	Cylinders
H8b, Pennsylvania Consolidation,	A	24 x 28"
H6b, Pennsylvania Consolidation,	A	22 x 28"
K2, Pennsylvania Pacific,	A	24 x 26"
E6, Pennsylvania Atlantic,	A	22 x 26"
E2d, Pennsylvania Atlantic,	A	20½x 26"
958, Ill. Central Consolidation,	B	22 x 30"
H6a, Pennsylvania Consolidation,	C	22 x 28"
734, L. S. & M. S. Consolidation,	C	21 x 30"
R, Chi. & N'w'n. Ten-wheel,	C	20 x 26"
Schenectady No.1, Purdue University, Eight-wheel,	C	17 x 24"
Same, No. 2,	C	16 x 24"
Same, No. 3, see note.		
E2a, Pennsylvania Atlantic	C	20½x 26"

Designation	Working Pressure	Diameter of Drivers	Theoretical T. E. (T_o)	Evaporation E
H8b (Cons.)	205 lb.	62"	53,400 lb.	46,000 lb.
H6b (Cond.)	205 lb.	56"	49,500 lb.	37,500 lb.
K2 (Pac.)	205 lb.	80"	38,400 lb.	65,000 lb.
E6 (Atl.)	205 lb.	80"	32,200 lb.	55,000 lb.
E2d (Atl.)	205 lb.	80"	28,000 lb.	43,000 lb.
958 (Cons.)	200 lb.	61"	47,600 lb.	38,000 lb.
H6a (Cons.)	205 lb.	56"	49,500 lb.	30,000 lb.
734 (Cons.)	200 lb.	63"	42,100 lb.	30,000 lb.
R (10-w.)	190 lb.	63"	31,400 lb.	31,500 lb.
S.#1(8-w.)	135 lb.	63"	14,900 lb.	17,000 lb.
S.#2(8-w.)	180 lb.	69"	15,900 lb.	21,000 lb.
E2a (Atl.)	205 lb.	80"	28,000 lb.	37,000 lb.

In addition to the engines listed in this table and Table 25, laboratory-test data are available for eight locomotives. Of these, five are compounds tested at St. Louis, and beyond the scope of this study. The engine

Schenectady No. 3 is a superheater, but is not comparable with the ordinary design of superheater locomotive on account of its having a Class C steam distribution system. No use is made of the data on account of its standing thus alone between the two classes. A Pennsylvania engine of the eight-wheel type (class D-16-d) has been tested at Altoona, but the range of speed was too limited to make the data of value for the present investigation. Engine 3929 of the New York Central was tested at Purdue University in 1906. This being a class B locomotive the data would be of special value in the present study, but no efficiency tests were run. The purposes of the tests were the study of front-end proportions and machine friction.

APPENDIX VI-B

METHODS OF USING AND MODIFYING THE DATA.

1. The H6b. This engine has been used on the Altoona testing-plant for several minor investigations, but no complete series of efficiency tests has been run. In Bulletin 13, reporting the tests of a smoke-box superheater, sufficient data are given for this engine to serve as a foundation for other assumptions as to its performance. In Bulletin 10 (p.95) is given a speed-pull curve for this engine, as obtained with the dynamometer-car in road-tests. From this curve a speed-horsepower curve has been plotted, and with it, the results of the laboratory tests in terms of speed and horsepower. Through the points representing the laboratory results a curve is of the same form as that developed in the road-tests, and the data used in Appendix VI-C are derived from this curve. (See Fig. A-3). The test-data are as follows:

TABLE 35, Data from Laboratory Tests, Engine H6b.

R. P. M.	M. P. H.	Piston Speed	I. H. P.
80	13.3	374	885
100	16.6	466	1157
120	20.0	560	1112

TABLE 36, Data from Road-tests, Engine H6b.

M. P. H.	Piston Speed	Drawbar Pull, lb.	Friction lb.	Cylinder T. E., lb	I. H. P.
0	0	40,800	4400	45,200	0
4	112	39,300	4400	43,700	466
8	224	34,300	4300	38,600	824
12	336	27,500	4200	31,700	955
16	448	22,000	4100	26,100	1112

To the values in the above table, that for 20 m.p.h. may be added by extrapolation, as the curve in Bulletin 10 extends to 19 m.p.h. At 20 m.p.h., the drawbar-pull is 17,500 lbs.; allowing 4000 lb. for friction and resistance, the cylinder tractive effort is 21,500 lb., and the indicated horsepower is 1145. The friction estimate used throughout is the same as that for the heavier H8b engine reduced in direct proportion to the weight on the drivers, and allowance is also made for the resistance of the tender and the trucks.

2. The H8b. As in the case of the H6b, it is necessary to supplement the laboratory data, as only two speeds are included at which normal tractive effort could be developed. The highest pulls at 60 and 80 revolutions per minute are lower than those developed at 100 revolutions. No information beyond that given in Bulletin 10 is available. Turning again to the road-test curves on p.95 it is found that at 18.4 m.p.h. , the only comparable point, the road-test cylinder tractive effort is 29,300 lb. On the testing-plant at this speed, 28,300 pounds was developed, three percent. less. In some degree this fact warrants the assumption that the rate of evaporation on the road was three percent. higher than the average maximum attained on the plant. The close check warrants the substitution of the road-test data using the value of E indicated in place of the 44,500 lb. maximum characteristic of the laboratory tests. The friction correction used is derived from the tests of the H8sb engine, similar as to weight and moving parts. The complete data as used appear

in Appendix VI-C.

3. The K2, E6, and E2d. The information for these engines is drawn from Bulletin 22. In each case the speed-range is limited, and in order to extend this, the dotted section of the curve, representing the estimate of the Test Department, is partially adopted. The curves for two of the engines are completed to zero-speed by the assumption that at the instant of starting, 98% of the boiler pressure is realized in the cylinders.

4. The 958. It is not possible to draw a speed-pull curve directly from the data as presented (in Bulletin 82 of the University of Illinois Engineering Experiment Station) as many of the high pulls represent very unusual firing rates. For this reason, a horsepower curve has been plotted, excluding those tests in which the firing rate was above 7000 lb. coal per hour. This curve represents almost the extreme limit of boiler performance. From it is taken the data for the speed-pull curve appearing in Appendix VI-C. The value used for "E" is that corresponding to a firing rate of 6000 lb. per hour, the average of the tests used. The horsepower curve is based upon the following tests:

TABLE 37, Indicated Horsepower, Engine 958.

Test. No.	Firing Rate lb. per hr.	M.P.H.	Piston Speed	Indicated Horsepower
2092	5491	30.6	840.8	1267
2095	3799	9.3	255.9	805
2098	----	9.4	258.2	822
2079	6332	42.1	1157.0	1109
2072	6802	19.9	546.8	1234
2074	6766	30.8	846.3	1264

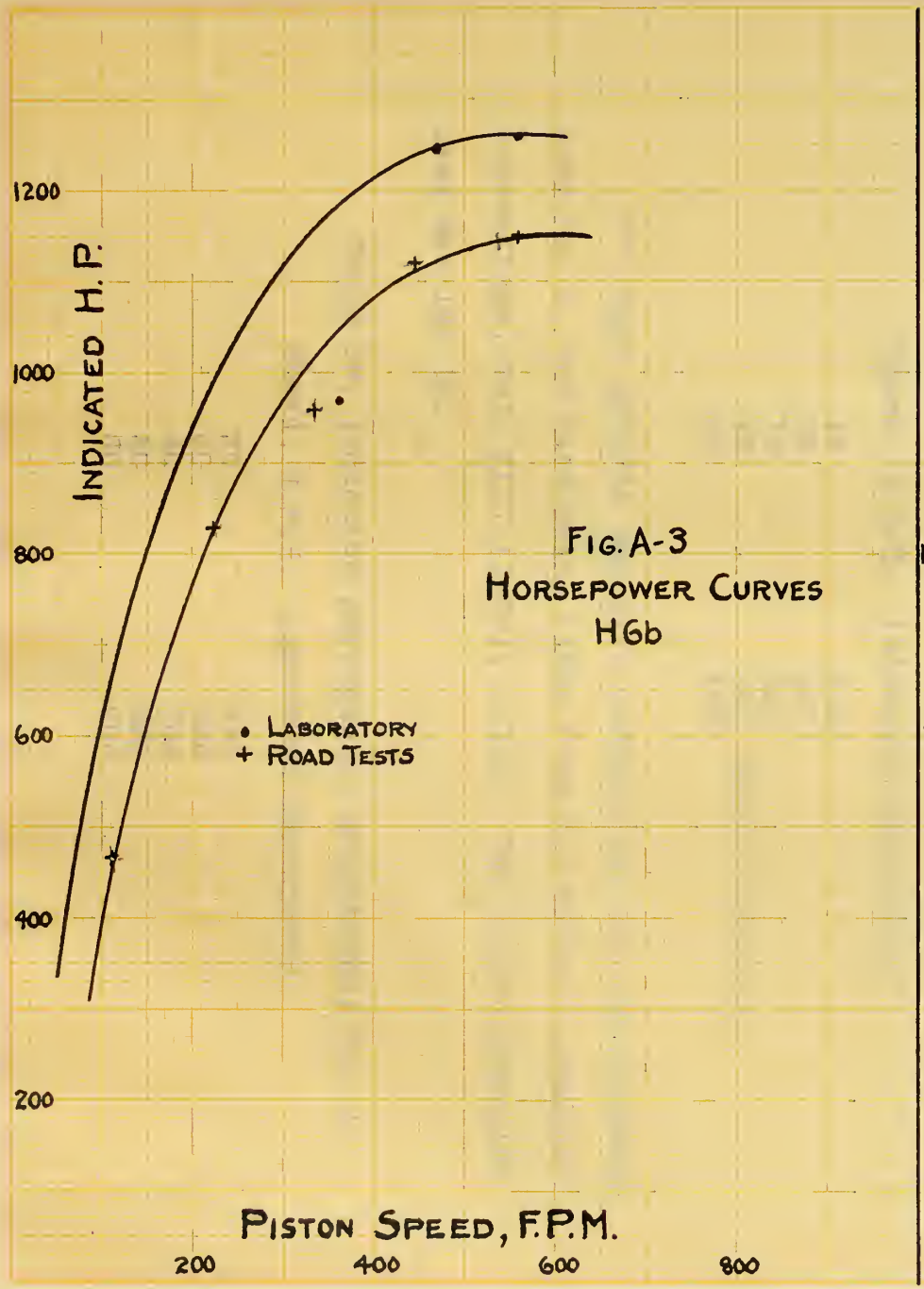


FIG. A-3
HORSEPOWER CURVES
H6b

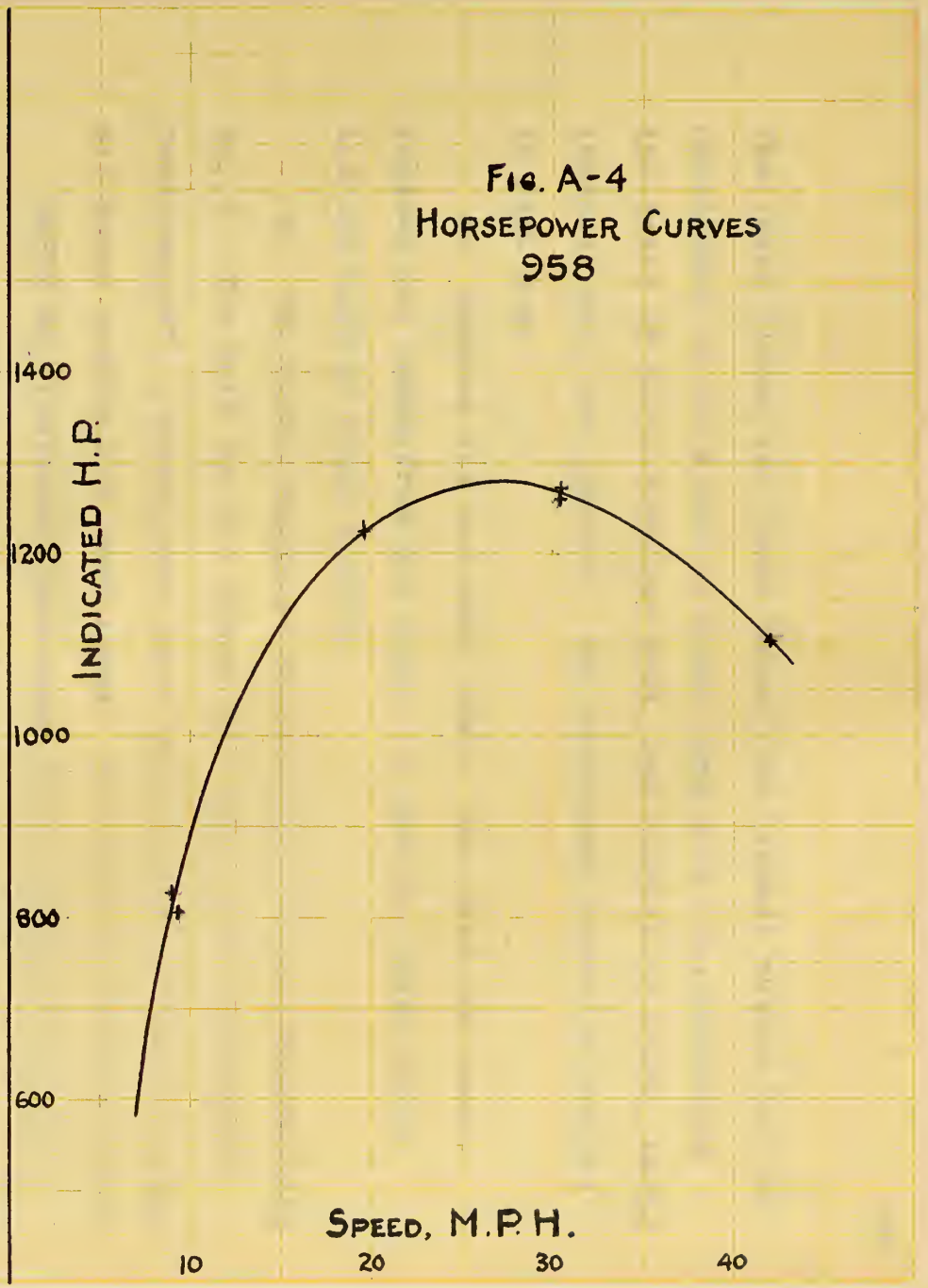
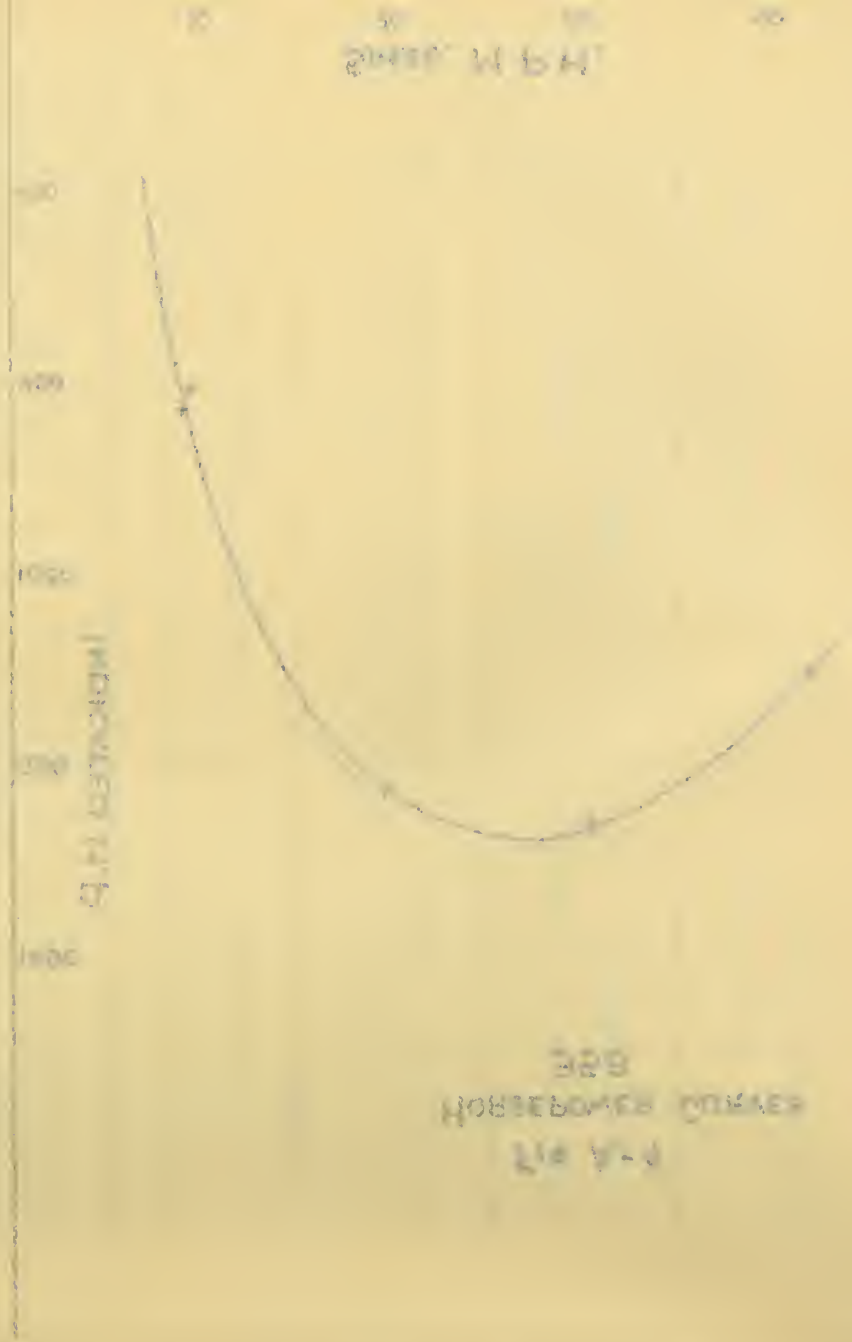


FIG. A-4
HORSEPOWER CURVES
958

INDICATED H.P.
BRAKE SPEED, R.P.M.
INDICATED
BRAKE


INDICATED H.P.
SPEED, R.P.M.

The tests listed are those having the highest values for the horsepower at the speed at which they were run, excepting the cases of tests 2076 and 2093, the results of which, being inconsistent, are ignored. The horsepower curve is plotted in Fig. A-4.

5. Engines 734 and 1499 (H6a). The data for these engines are taken from the report of the St. Louis tests, and are used as found therein.

6. The Purdue Engines--Schenectady No.1 and Schenectady No. 2. The data for the former are taken from "Locomotive Performance", a horsepower curve being plotted from the highest power produced at each speed. The figures for this curve:

TABLE 38, Indicated Horsepower Curve, Schenectady No.1.

Speed, M.P.H.	Indicated Horsepower
15	319
25	451
35	522
45	438 (Below capacity)
55	438

The same process has been carried out for the Schenectady No. 2, the data being taken from the Master Mechanics' Proceedings, Vol. XLII (1909), p.162. The boiler pressure used was 180 lb.

TABLE 39, Indicated Horsepower, Schenectady No. 2.

Speed, M.P.H.	Indicated Horsepower
20	495
30	664
40	762
50	692
60	600

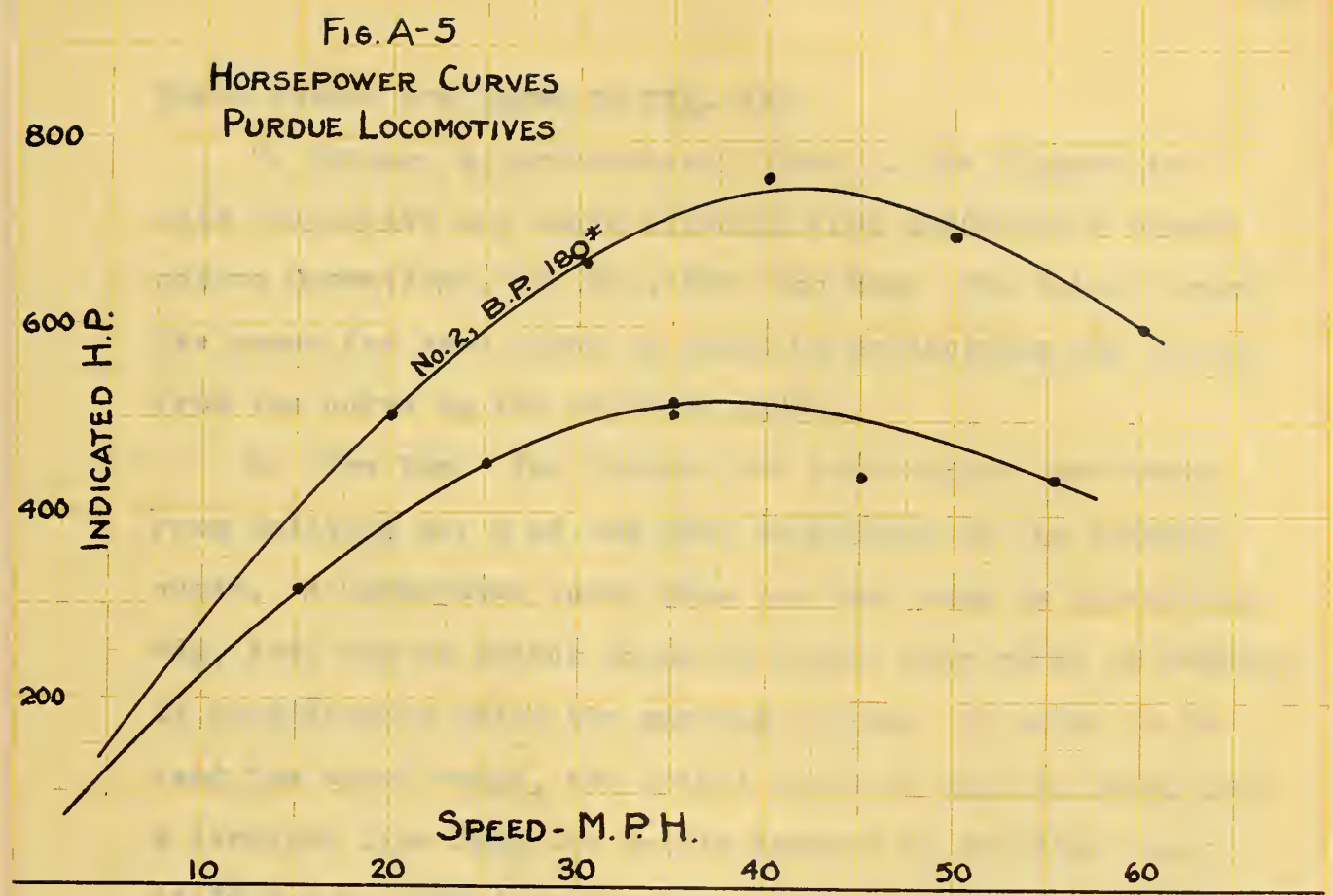
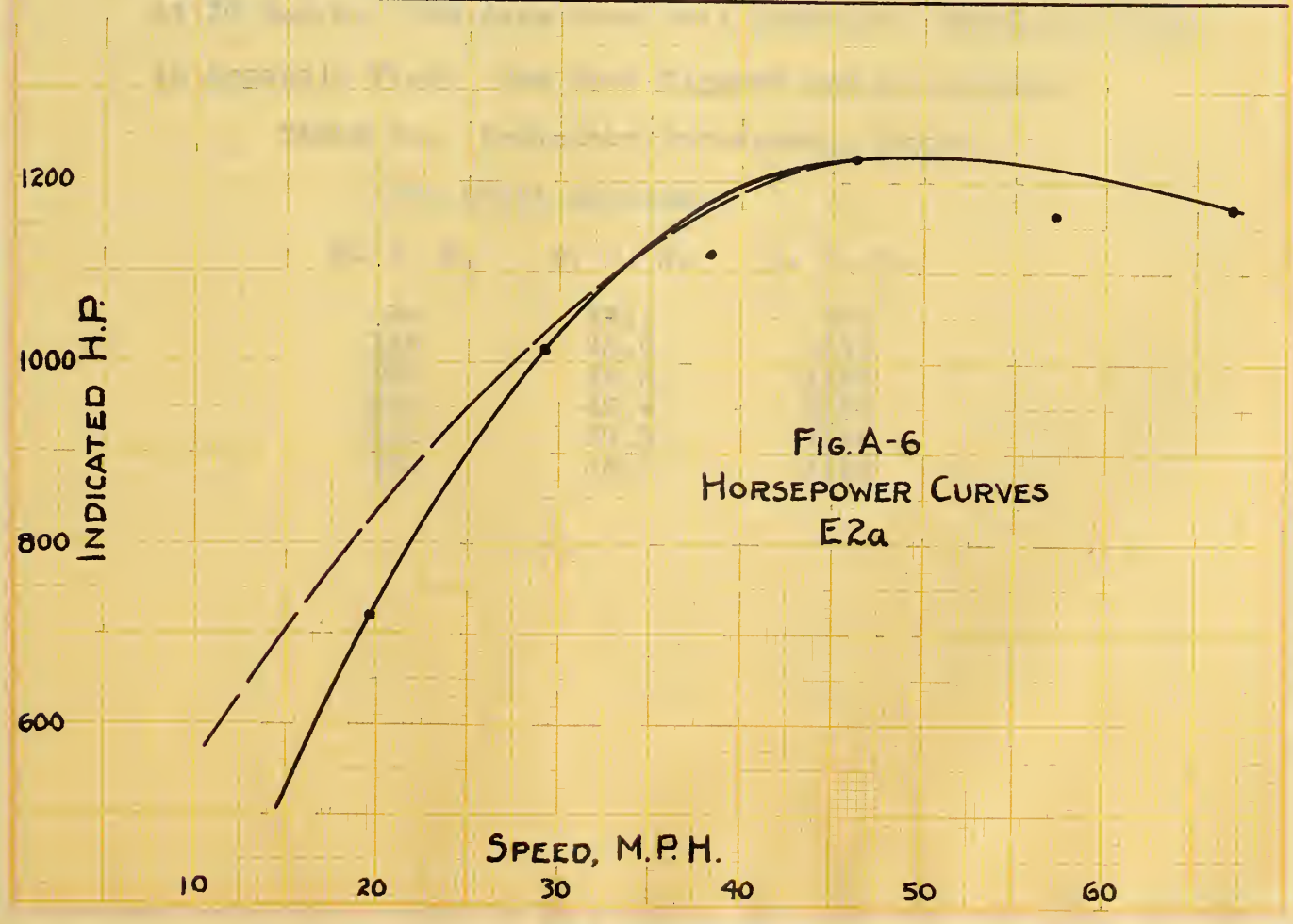
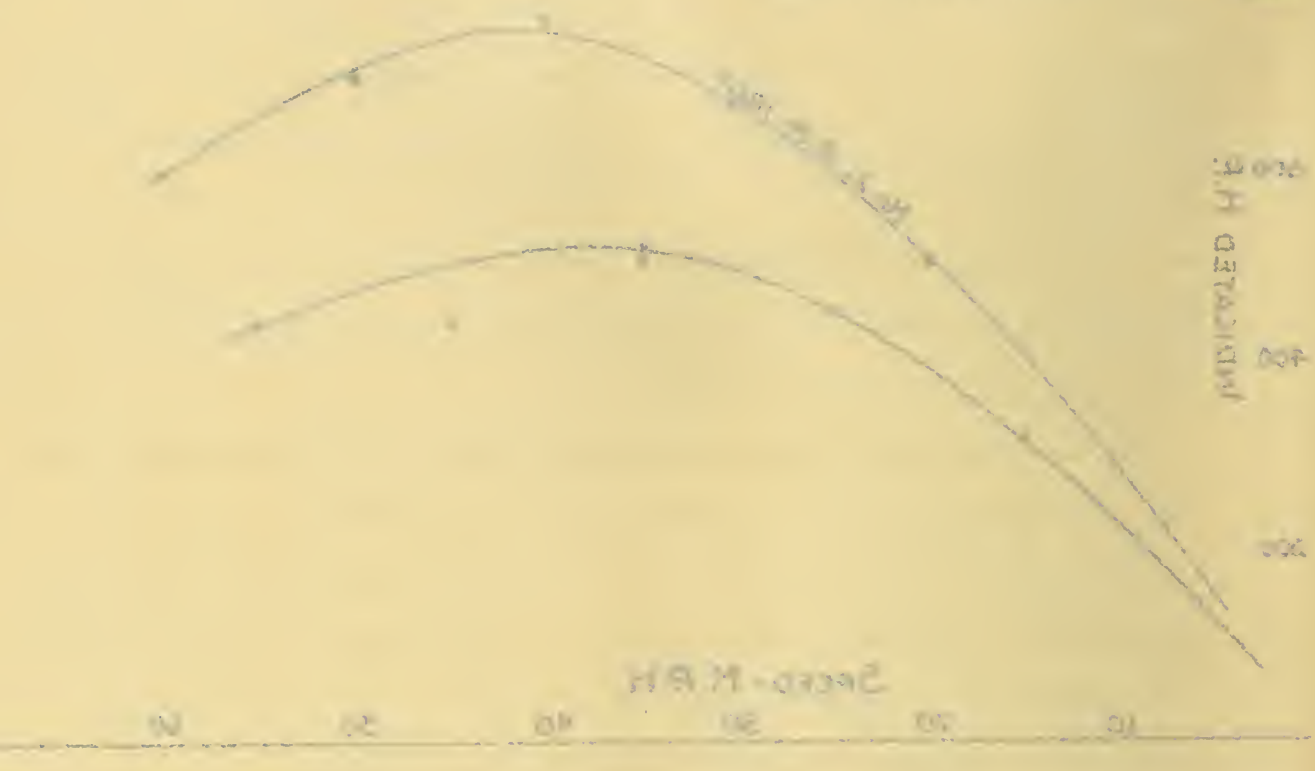
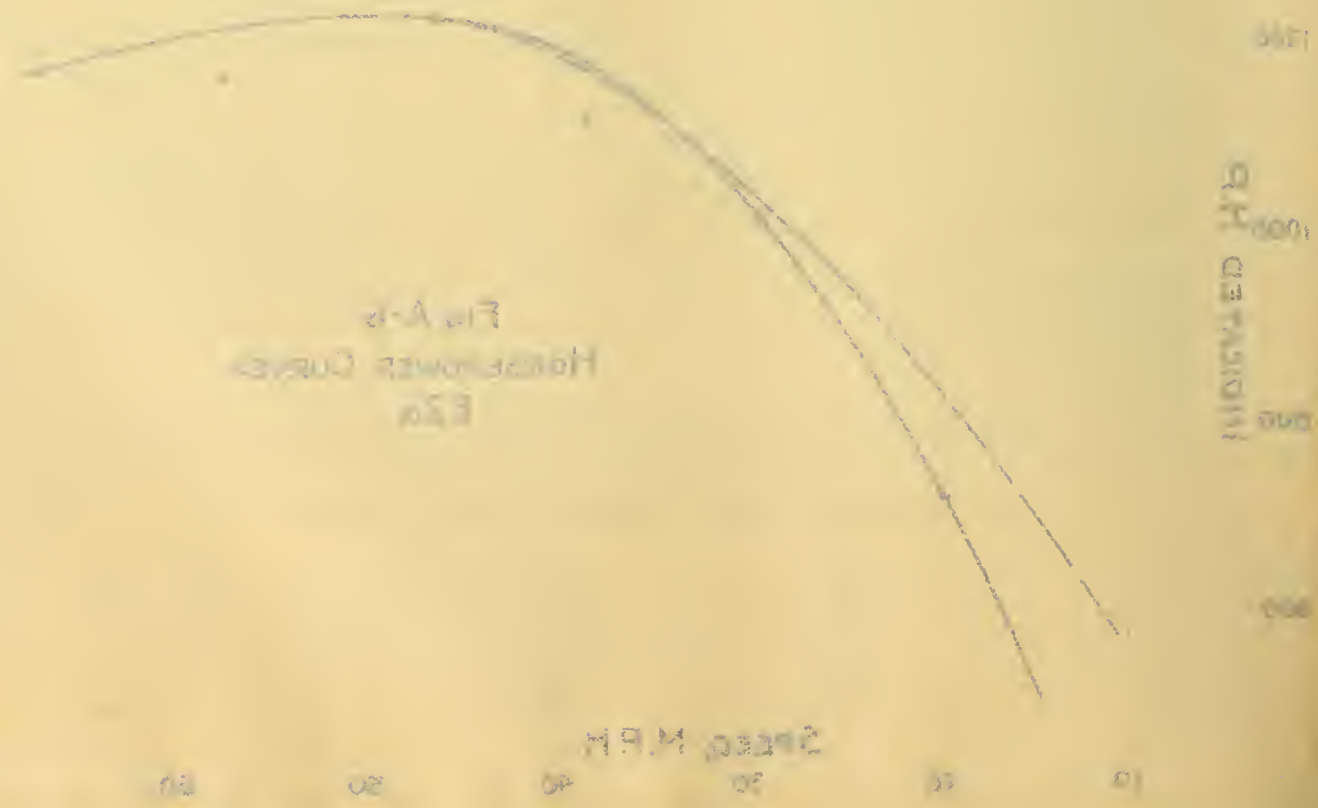


Fig. A-5
Horsepower Curves
Purdue Locomotives
800
600
400
200
Indicated H.P.
No. 2, B.P. 180#
Speed - M.P.H.
10
20
30
40
50
60


1200
1000
800
600
Indicated H.P.
Fig. A-6
Horsepower Curves
E2a
Speed, M.P.H.
10
20
30
40
50
60


Speed - M.P.H.
10
20
30
40
50
60
Indicated H.P.
300
400
500
600


Speed, M.P.H.
10
20
30
40
50
60
Indicated H.P.
800
900
1000
1100

These curves are shown in Fig. A-5.

7. Chicago & Northwestern, Class R. The figures for this locomotive are taken directly from Henderson's "Locomotive Operation", pp. 351, 359, and 362. The actual tractive power for each speed is found by multiplying the factor from the curve by the value of pld^2/D.

8. The E2a. The figures for this engine were taken from Bulletin No. 5 of the Test Department of the Pennsylvania. A horsepower curve from the test data is plotted in Fig. A-6, but at speeds below 30 m.p.h. this curve is evidently considerably below the maximum output. In order to extend the speed range, the dotted curve is run in, based upon a straight line from the origin tangent to the test curve at 30 m.p.h. The data from this composite curve are given in Appendix VI-C. The test figures are as follows:

TABLE 40, Indicated Horsepower, Engine E2a.(Test Results).

R. P. M.	M. P. H.	I. H. P.
80	19.1	727
120	28.6	1015
160	38.2	1133
200	46.4	1223
240	57.3	1164
280	66.7	1178

APPENDIX VI-C

TABLE 41, Data for Speed-pull Curves, Locomotives Using Saturated Steam.

Miles per Hour	Piston Speed	I.H.P.	Drawbar Pull, lb.	Friction lb.	Cylinder T.E., lb.	Ratio C.T.E./E
H8b Consolidation, T_o= 53,300, E = 46,000, T_o/E = 1.16						
0	0	-	45,700	4900	50,600	1.100
5	126	-	42,900	4800	47,400	1.040
8	202	-	39,800	4600	44,400	.962
10	252	-	37,200	4400	41,600	.905
15	378	-	29,900	4200	34,100	.740
20	504	-	18,900	3700	22,600	.490
H6b Consolidation, T_o= 49,600, E = 37,500, T_o/E = 1.32						
7.5	210	920	-	-	46,000	1.270
12.0	336	1155	-	-	36,200	.955
16.7	466	1248	-	-	28,000	.750
20.0	560	1258	-	-	23,500	.625
22.5	630	1260	-	-	21,000	.560
K2 Pacific, T_o= 38,400, E = 65,000, T_o/E = 0.580						
10	182	-	33,300	3100	36,400	0.560
20	364	-	28,000	3000	31,000	.476
30	546	-	22,000	2900	24,900	.384
40	728	-	17,000	2800	19,800	.302
E2d Atlantic, T_o= 28,000, E = 43,000, T_o/E = 0.650						
10	182	-	23,300	2000	25,300	0.590
20	364	-	19,000	2000	21,000	.490
30	546	-	14,600	2000	16,600	.390
40	728	-	11,000	2200	13,200	.310
50	910	-	8,400	2400	10,800	.252
E6 Atlantic, T_o= 32,200, E= 55,000, T_o/E = 0.585						
25.0	474	-	20,750	2300	24,050	0.440
30.0	546	-	18,400	2300	20,700	.378
37.5	684	-	15,500	2300	17,800	.324
45.0	822	-	12,600	2300	24,900	.274

TABLE 41, Continued.

(The fourth and fifth columns are not required for the remaining data, and are therefore omitted.)

Miles per Hour	Piston Speed	I.H.P.	Cylinder T.E.,lb.	Ratio C.T.E./E
958 Consolidation, T_o= 47,600, E = 38,000, T_o/E = 1.280				
10.0	276	870	32,600	0.860
15.0	412	1120	28,000	.738
22.5	620	1258	20.900	.548
30.0	825	1260	15,800	.415
37.5	1030	1180	11,800	.310
45.0	1236	1025	8,750	.232
H6a Consolidation, T_o= 49,600, E= 30,000, T_o/E = 1.687				
6.7	187	715	40,400	1.346
13.3	374	931	26,200	.873
20.0	560	1036	19,450	.648
26.7	747	1024	15,000	.500
734 Consolidation, T_o= 42,100, E = 30,000, T_o/E = 1.403				
7.5	200	738	36,900	1.230
15.0	400	962	24,050	.801
30.0	800	1054	13,150	.405
Schen.#1, 8-wheel, T_o= 14,900, E = 17,000, T_o/E = 0.875				
10.0	213	228	8,550	0.502
15.0	320	320	8,000	.470
22.5	480	420	7,000	.411
30.0	640	498	6,200	.365
45.0	960	498	4,160	.245
60.0	1280	400	2,500	.147
Schen.#2, 8-wheel, T_o= 15,950, E = 21,000, T_o/E = 0.760				
10.0	194	280	10,500	0.500
15.0	291	400	10,000	.475
25.0	486	600	9,000	.429
37.5	680	740	7,400	.352
50.0	975	700	5,250	.250
60.0	1170	600	3,750	.178

TABLE 41, Continued.

Miles per Hour	Piston Speed	I. H. P.	Cylinder T.E., lb.	Ratio C.T.E./ E
Class R, 10-wheel, T_o= 31,400, E = 31,500, To/E = 1.000				
3.75	86	-	27,600	0.876
7.50	172	-	27,200	.864
11.25	260	-	26,600	.845
15.00	346	-	23,900	.756
18.75	433	-	19,150	.609
22.50	520	-	16,350	.519
30.00	602	-	12,900	.409
37.50	867	-	10,000	.317
45.00	1040	-	7,300	.232
E2a Atlantic, T_o/E = 0.750, T_o= 28,000, E = 37,000.				
15.00	273	700	17,500	0.465
20.00	364	830	15,550	.414
30.00	546	1050	13,100	.350
37.50	683	1175	11,750	.313
50.00	910	1230	9,220	.246
60.00	1092	1210	7,560	.202

The first five engines in this table have Class A steam distribution, and are plotted in Fig. 15, as is also the 958, of Class B. The remaining six, of Class C, are shown in Fig. 16.

APPENDIX VI-D

TABLE 42, Table of Ordinates, Figs. 15 and 16.

Engine Designation	Piston Speed, f.p.m. 100	200	300	400	600	800	1000	1200
Ordinates as read from curves, Figs. 15 and 16:								
H6b	1.260	1.200	1.030	0.840	0.590	-	-	-
H8b	1.050	.970	.840	.710	.510	-	-	-
K2	.560	.550	.510	.450	.350	0.270	-	-
E6	-	-	-	-	.350	.270	0.230	-
E2d	.660	.600	.530	.470	.370	.290	.240	-
958	1.210	1.100	.960	.750	.565	.425	.320	0.240
H6a	1.600	1.300	1.030	.840	.610	.465	-	-
734	1.370	1.230	.980	.800	.560	.405	-	-
R	.875	.855	.780	.660	.445	.340	.260	.200
Sch.#1	-	-	-	-	.365	.295	.230	.170
Sch.#2	-	-	-	-	.380	.310	.240	.170
E2a	-	-	.530	.450	.330	.285	.220	.180
The same, in fractions of the ordinate at 600 f.p.m.:								
H6b	2.140	2.030	1.750	1.410	1.000	-	-	-
H8b	1.980	1.900	1.650	1.390	1.000	-	-	-
K2	1.600	1.570	1.460	1.290	1.000	.775	-	-
E6	-	-	-	-	1.000	.775	.660	-
E2d	1.780	1.620	1.430	1.260	1.000	.785	.650	-
958	2.140	1.960	1.700	1.330	1.000	.750	.565	.425
H6a	2.650	2.100	1.700	1.380	1.000	.760	-	-
734	2.450	2.200	1.750	1.430	1.000	.725	-	-
R	1.970	1.920	1.800	1.480	1.000	.760	.584	.450
Sch.#1	-	-	-	-	1.000	.810	.630	.470
Sch.#2	-	-	-	-	1.000	.815	.630	.450
E2a	-	-	1.630	1.360	1.000	.860	.665	.545

APPENDIX VI-E

TABLE 43, Relation Between Tractive Effort at 600 f.p.m., T_o, and Boiler Capacity.

Engine	T_o, lb.	E, lb.	T_o/E	t/E	t/T_o
H6b	49,600	37,500	1.320	0.590	0.447
H8b	53,400	46,000	1.160	.510	.440
K2	38,400	65,000	.580	.350	.604
E6	32,200	55,000	.585	.350	.600
E2d	28,000	43,000	.650	.370	.570
958	47,600	38,000	1.280	.565	.441
H6a	49,600	30,000	1.690	.610	.360
734	42,000	30,000	1.403	.570	.406
R	31,400	31,500	1.000	.460	.460
Schen.1	14,900	17,000	.875	.365	.424
Schen.2	15,900	21,000	.760	.380	.500
E2a	28,000	37,000	.750	.330	.454

For convenience, the meanings of the symbols used are here repeated:

T_o = the theoretical tractive effort, or pld^2/D.

E = the normal maximum hourly equivalent evaporation, depending upon the design of the boiler, and the quantity and quality of fuel fired.

t = the cylinder tractive effort in pounds at 600 f.p.m. piston speed.

APPENDIX VII-A

EVAPORATION ESTIMATES BY VARIOUS METHODS.

In the paragraphs to follow, the evaporation of a boiler of known performance is estimated by several theoretical methods. The estimates are based on the boiler of the Pennsylvania E2d Atlantic, tested at Altoona. The dimensions are as follows:

Tube Heating Surface, water-side,	2288 sq. ft.
Firebox Heating Surface,	162 sq. ft.
Total Heating Surface,	2450 sq. ft.
Grate Area	55.3 sq. ft.
Length of Tubes,	15 ft.
Diameter of Tubes, and Spacing	2 in., 5/8 in.

From the tests, when the firing rate was 100 lb. coal per sq. ft. of grate per hour, the equivalent evaporation was 42,600 lb. steam per hour. This corresponds to an actual evaporation of 35,500 lb. per hour. The coal used had a thermal value of 14,500 B.t.u. The firing rate named (5530 lb. coal per hour) will be used in the cases where the coal rate is considered.

1. The Cole Estimate. For the given tube dimensions, the rate of evaporation is 9.44 lb. per sq. ft. of h. s. per hour. The total evaporation per hour (actual) is then

9.44 x 2288	=	23,300 lb.	(tubes)
162 x 55	=	8,900 lb.	(firebox)
		30,800 lb.	(total)

2. Houston's Estimate. Coal of 14,500 B.t.u. should evaporate 12.60 lb. of water (equivalent) per sq. ft. h. s. per hour, provided the coal is fired at the rate of $1\frac{3}{4}$ lb. per sq. ft. of h. s. per hour. The latter being true (5530 ÷ 2450 = 2.26) the hourly equivalent evaporation is 2450 x

12.60 lb. or 30,800 lb. equivalent steam per hour. If it be considered that an excess of coal above 1.75 lb. per square foot per hour will cause a proportionally larger evaporation, the estimate should be multiplied by the ratio 2.26/1.75, giving an hourly equivalent evaporation of 39,500 pounds.

3. Williamson's Estimate. Each thousand heat units fired should evaporate 0.68 lb. of water, when the firing rate is 1 lb. per sq. ft. of h. s. per hour. The coal has a thermal value of 14,500 B.t.u., hence the estimated evaporation is 14.5 x 2450 x 0.68 = 24,000 lb. actual evaporation per hour. This estimate is scarcely comparable with the others on account of the assumption of such a low firing rate. At this rate (2450 lb. per hour) the actual evaporation of the E2d boiler was 21,000 lb. per hour.

4. Henderson's Estimate. Assume that the coal used had 60% of fixed carbon, and 9% of noncombustible matter, representative figures for Pennsylvania coal such as was used in these tests. The evaporation rate from Mr. Henderson's chart for this composition is 10.15 lb. water per lb. coal, and the total equivalent evaporation per hour is then 5530 x 10.15 = 56,200 lb.

5. Shurtleff's Original Method. For this engine H.S./ G.A. is 44; for this ratio and the assumed firing rate the evaporation is 14.7 lb. water per sq. ft. h. s. per hour. The total is 14.7 x 2450 = 36,000 lb. equivalent steam per hour.

6. The A. R. E. A. Method. For a firing rate of 2.26 lb. per sq. ft. h. s. per hour, the coal having 14,500 B.t.u., the evaporation per lb. of coal is 4.75 lb. actual steam, and the

total per hour is 5530 x 4.75 = 26,300 lb.

TABLE 44, Summary of Evaporation Estimates.

Method of Estimating	Evaporation per hour, lb. Actual	Equivalent	Firing Rate lb. per hr.
Tests	42,600	35,500	5530
A.R.E.A.	31,600	26,300	5530
Henderson	56,200	47,000	5530
Cole	38,600	32,200	Indefinite
Shurtleff	36,000	30,000	5530
Tests	37,500	31,250	4300
Houston	30,800	25,700	4300
Tests	25,200	21,000	2450
Williamson	29,000	24,000	2450

APPENDIX VII-B

TABLE 45, Mahler's Calorimetric Tests.*

Fixed Carbon, Percent. of Total Combustible	Heat Value of Combustible, B.t.u.	Equiv. Evap. per lb. of Combustible, lb.
80	15,800	16.40
72	15,660	16.21
68	15,480	16.03
60	14,580	15.09
54	13,320	13.79
50	12,240	12.67

The partial analysis given above may be completed by analogy--the comparison of similar coals, and the last two columns of the table modified to terms of "coal as fired" in place of "combustible". The following table carries out the process, the six coals given corresponding to the six

* Kent's M. E. Pocket-book (1906 ed.) p. 634.

given in the Mahler table.

TABLE 46, Modification and Completion of the Mahler Table.

Analysis of Coal Fixed Carbon %	Volatile %	Ash & Moisture %	B.t.u. in Coal as Fired	Equ. Evap. per lb. Coal as Fired, lb.
75	17	8	14,500	15.2
65	25	10	14,100	14.6
59	28	13	13,500	14.0
50	33	17	12,100	12.5
43	36	21	10,500	10.9
38	37	25	10,000	9.5

In adapting the data of Mr. Bement* to the purpose of the present investigation, it will be assumed that the coal used (mined in Williamson County, Illinois) had a thermal value as fired of 11,000 B.t.u., and that the ash mixed with it contained 3,000 heat units. The curve drawn by Mr. Bement is unsatisfactory, and the points given are taken from a more representative locus.

TABLE 47, Bement's Fuel Tests

Percent. of Ash Mixed With Coal	Efficiency of Boiler Percent.	B.t.u. in 1 lb. of Mixture In Coal	In Ash	Total	Equ.Evap. per lb.of Mixture
0	60	11,000	0	11,000	6.8
10	59	9,900	300	10,200	6.2
20	57	8,800	600	9,400	5.5
30	52	7,700	900	8,600	4.6
35	40	7,150	1,050	8,200	3.4
40	0	6,600	1,200	7,800	0

The last column is derived from the efficiency of the boiler.

---

* Proc. Western Railway Club, April 20, 1909.

Seventy-eight tests of almost as many varieties of coal were made by the U. S. Geological Survey at St. Louis in 1904, all of the tests having been made in two identical boilers. The report of the tests (Technical Paper No. 48, pt.II) gives all of the information necessary to make the data applicable to the present purpose. In each case the thermal value of the dry coal is given, and to render the results comparable with those developed from the Mahler and Bement tests, the B.t.u. value of the coal is calculated as fired . This is done by multiplying the B.t.u. of the dry coal by the factor 1.0 -- (the percent. of moisture)/100. The test data are very consistent with a mean curve through the points. Of the 78 tests 39 have their points on the mean curve or with one percent.or less variation; 20 more vary from the mean curve between one and five percent.; 15 vary between five and ten percent., and in four cases only is the variation greater than the latter figure.

TABLE 48, U. S. G. S. Boiler Tests.

No.Test	Name of Coal	B.t.u. of Dry Coal	Moisture,%	B.t.u. Coal as Fired	Equ.Evap. per lb. Coal
22	Texas Lignite	10,900	23.27	8,400	3.60
63	Wyoming Lignite #3	12,300	21.81	9,600	5.47
48	Illinois #4	12,400	13.47	10,700	6.38
1	Pennsylvania #1	14,700	1.10	14,500	8.94
16	Alabama #2	12,600	4.83	12,000	8.13
19	Illinois #2 washed	12,600	11.45	11,300	7.16
13	Kansas #2 washed	14,000	5.82	13,200	8.38
59	W. Va. #12,briquette	15,200	1.58	14,900	9.97

APPENDIX VII-C

TABLE 49, Evaporation Data from Coateville Tests, Series A.

Test No.	Dry Coal per hr. lb.	per sq.ft.	Equivalent Evaporation per sq.ft. h.s. per hour, lb. Firebox	Total h.s.	per # Coal	Ratio of Evaporations
A-101-J	1324	22.8	26.64	4.82		5.48
A-102-J	1622	27.8	30.09	4.74		6.25
A-103-J	2498	43.0	38.77	8.42		4.60**
A-104-J	2990	51.4	42.57	8.54		4.98**
A-105-J	4228	72.6	51.92	11.77		4.41**
A-106-J	4392	75.5	52.18	11.43		4.55**
A-107-R	1721	29.6	34.95	4.95		7.05
A-108-R	1434	24.6	30.88	5.03		6.15
A-109-R	2418	41.5	37.42	8.10		4.60**
A-110-R	2835	48.7	47.64	8.19		5.75**
A-111-R	4250	73.0	58.02	11.99		4.83**
A-112-R	4138	71.1	53.26	11.58		4.60**

The dimensions of the two boilers were as follows:

Total Heating Surface	2996 sq.ft.
Firebox Heating Surface(average of two)	220 sq.ft.
Grate Area	58.1 sq.ft.

The average value of the eight ratios marked thus ** is 4.78. These ratios are for the tests having a firing rate above 2000 lb. of coal per hour, rates below this having no importance when operating conditions are considered, bearing in mind the size of the boilers.

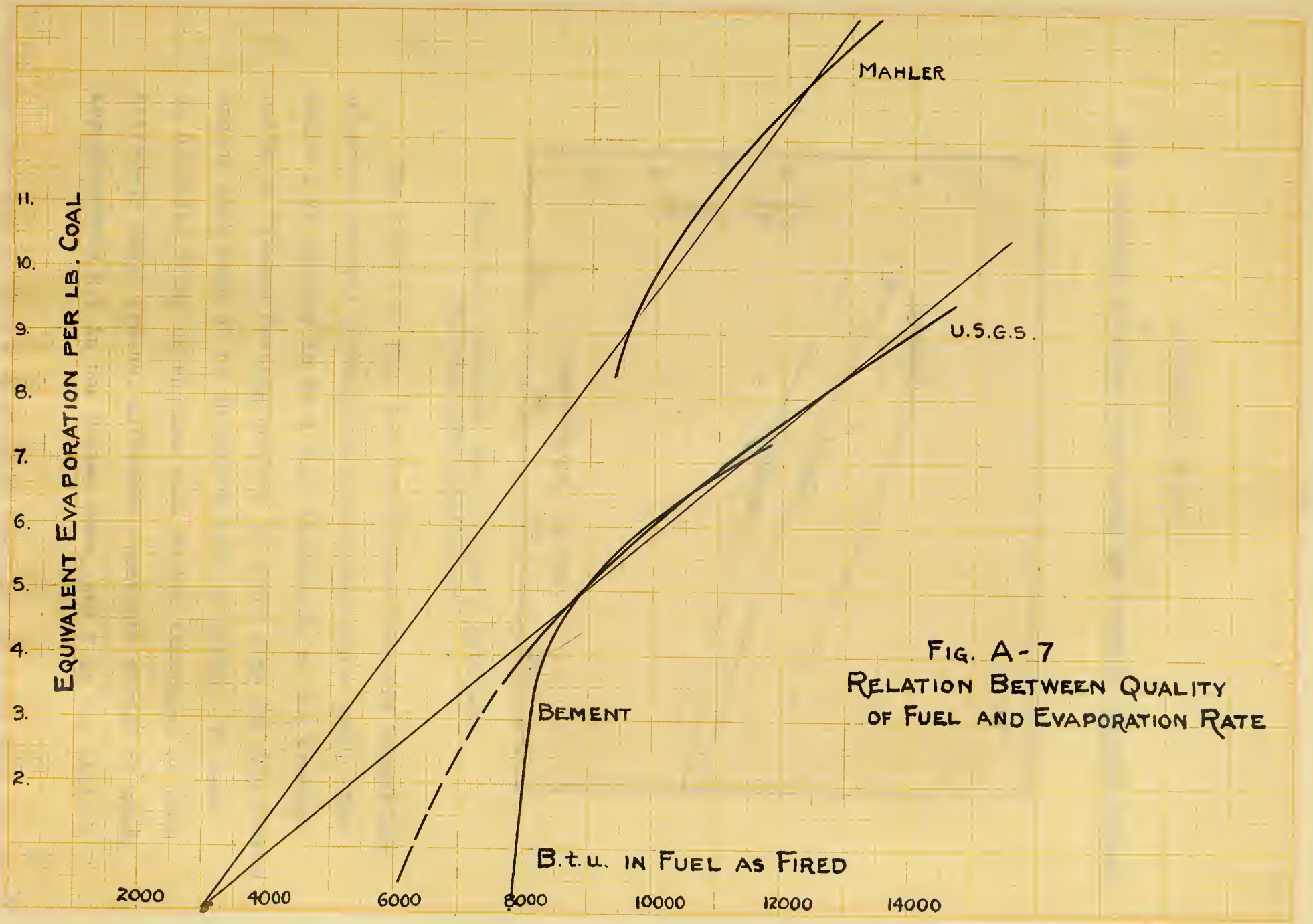
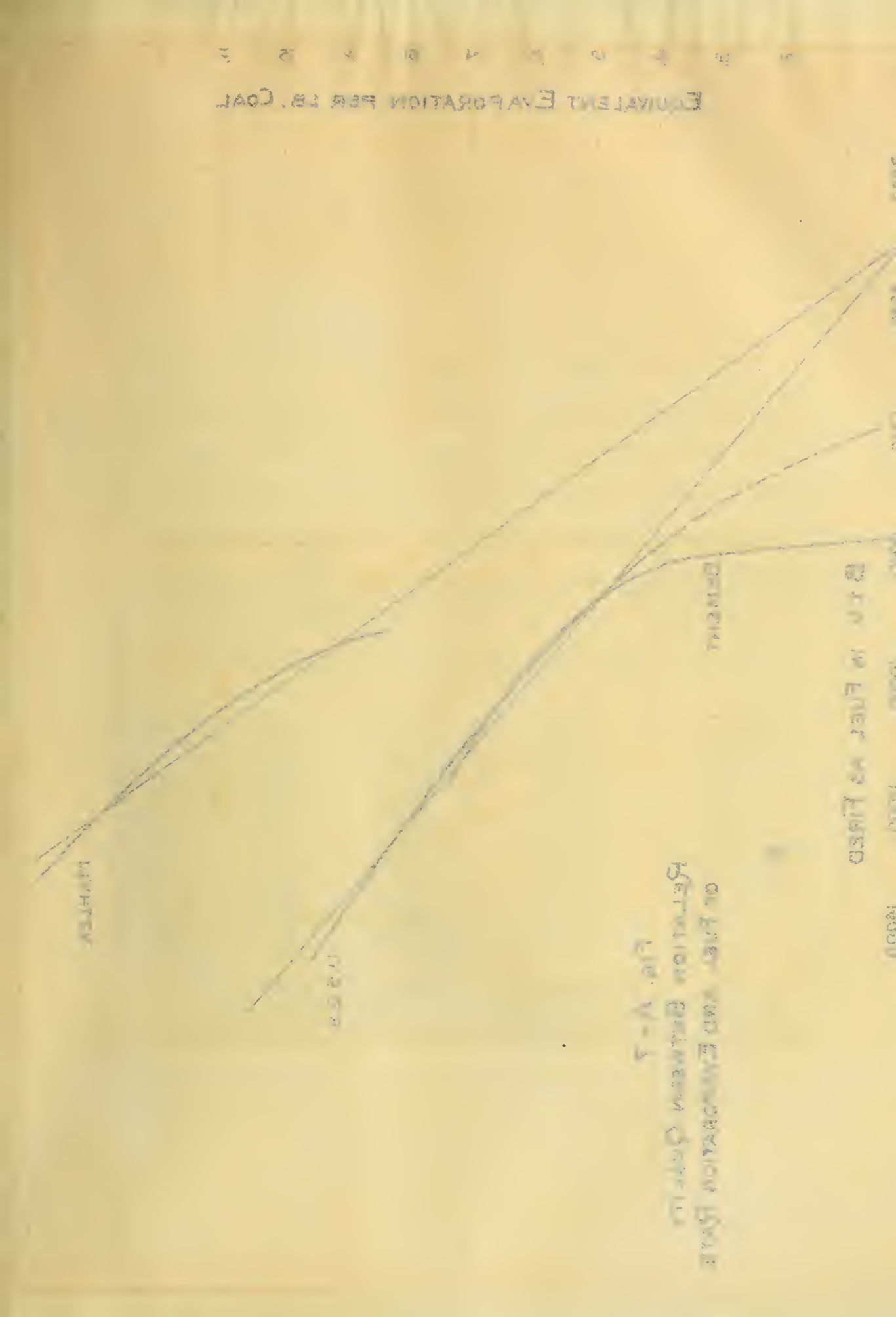


FIG. A-7
RELATION BETWEEN QUALITY OF FUEL AND EVAPORATION RATE



APPENDIX VII-D

THE RELATION BETWEEN GRATE AREA AND FIREBOX HEATING SURFACE.

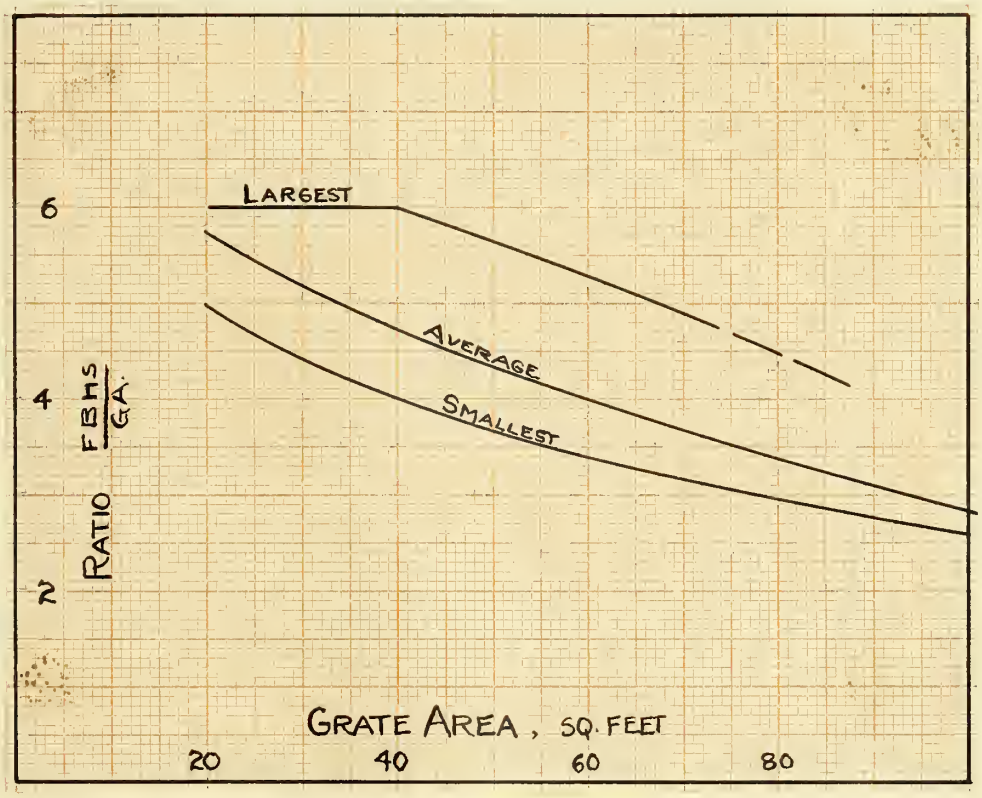


Fig. A-8, Relation Between Grate Area and Firebox Heating Surface.

The curves in Fig. A-8 are plotted from the dimensions of about seventy locomotives constructed since 1900. The value 3.75 appears to be a fair assumption as the average ratio of firebox heating surface to grate area for the modern engine with from 40 to 80 square feet of grate. To make an accurate study of engines with Wooten fireboxes or with extremely small grates, different constants should be used: approximately 2.5 in the former case, and 6 in the latter.

APPENDIX VII-E

DATA FOR EVAPORATION CURVES.

There are available from various sources reports of tests of twenty-seven locomotive boilers. These tests are as follows:

On the Purdue testing-plant, three engines, reported in "Locomotive Performance" (Goss) and the Master Mechanics' Proceedings for 1908.

On the Pennsylvania plant at St. Louis, eight engines, reported in "Locomotive Tests and Exhibits", published by the Pennsylvania Railroad.

On the Pennsylvania plant at Altoona, fourteen engines, the records being given in the bulletins of the Test Department of the Pennsylvania Railroad.

On the University of Illinois plant, one engine, reported in Bulletin 82 of the U.of I. Engineering Experiment Station.

At Coatesville, Pa., two boilers , considered as one as they were identical in heating surface characteristics. The tests are reported in a volume published by the Jacobs-Shupert United States Firebox Company.

The heating surface details for all of these engines are given in Table 50. In most cases the report of the tests gives a curve showing the relation between coal fired per sq. ft. of grate surface per hour, and equivalent evaporation per pound of coal. In such cases, knowing the firing rate in terms of the grate area, the evaporation rate can be read

directly. In the remaining cases it was necessary to compute the total evaporation and divide this by the coal fired. When the latter step was necessary, the total evaporation is shown in the tables which follow.

Table 51 presents the data upon which Fig. 20 is based, and the original plot of these data is shown in Fig. A-9. The curves (in Fig. 20) for firing rates of 12,16,20,22, and 24 lb. coal per square foot of E.G. per hour are interpolated between the curves developed in Fig. A-9.

It should be noted that all of the engines except two were tested with coal averaging around 14,500 B.t.u. In the other cases (the 958 and Schenectady No.1) the evaporation rates as obtained in the tests are increased by the appropriate coal-factor.

APPENDIX VII-E cont.

TABLE 50, Heating Surface Data for All Engines.

Engine Designation	Heating Surface* sq.ft.	Superh'g Surface sq. ft.	Grate Area sq.ft.	Tube Length ft.	Equivalent Grate First Formula‡	% Gain	Equivalent Grate Second Formula‡
Sch.#1	1347	- -	17.3	11.5	92.1	1	92.1
Sch.#2	1350	- -	17.0	11.5	92.1	15.5	92.1
Sch.#3	1040	200	17.0	11.5	74.8	26.7	85.9
628 **	1618	284	29.1	12.0	118.6	24.5	134.9
H6a	2843	- -	49.2	13.7	207.1	23.7	207.1
H6b	2840	- -	48.7	13.7	206.4	23.5	206.4
E6s(old)	2659	689	55.2	13.7	202.7	26.4	241.2
E6	3619	- -	54.7	14.0	256.2	21.4	256.2
734	2858	- -	33.8	14.9	190.8	17.1	190.8
2512	1646	- -	33.3	14.7	124.8	[illegible]	124.8
E2a	2626	- -	55.5	15.0	201.5	27.1	201.5
E3sd	2105	561	54.7	15.0	171.7	31.5	202.7
E6s(new)	2867	811	55.8	15.0	214.9		260.0
H8sb	3030	809	55.3	15.0	223.3		268.3
H8b	3840	- -	55.3	15.0	268.5		268.5
E2d	2446	- -	55.3	15.0	191.3		191.3
3000	3407	- -	49.9	16.0	239.8		239.8
585	3181	- -	49.4	16.0	226.3		226.3
958	3688	- -	49.6	16.3	254.2		254.2
Coates.	2996	- -	58.1	18.2	224.6		224.6
K4s	4035	1171	69.3	19.0	294.1		359.3
929	4817	- -	58.4	19.9	326.4		326.4
535	3237	- -	48.4	18.8	228.0		228.0
K2as	3644	990	53.2	20.9	280.0		310.7
K2as(str)	3670	990	54.1¶	20.9	269.7		300.4
K2	4619	- -	55.4	20.9	312.0		312.0
K29s	4606	1302	70.7¶	21.9	326.7		386.0

* Water-side of tubes, superheating surface not included.

** Data for the most part unsatisfactory, due to the unsuitability of the coal used. Designed for German coal.

¶ The grates of these two engines were taken as 41.4 and 58.0 sq. ft., these being the respective values of the active grates. The stoker retorts reduce the effective area.

‡ The second formula is the one used in this investigation, i.e., E.G.= (H.S.+Sup.S.)/18+G.A. The first formula, as used in some of the preliminary investigations, omits the superheating surface from the formula.

TABLE 51

Schen. #1	92.1	2030	117.0	—	7.70	11.5	
" #2	92.1	2030	117.0	—	8.40	11.5	
" #3	85.9	1890	111.0	—	8.80	11.5	
628	134.9	2970	103.0	21,700	7.30	12.0	
H6a	207.1	4550	93.0	29,300	6.50	13.7	
H6b	206.4	4560	93.0	—	7.80	13.8	
E6s(old)	241.2	5300	96.0	—	8.60	13.7	
E6	256.2	5610	102.0	—	8.80	14.0	
734	190.8	4200	124.0	30,000	7.10	14.9	
2512	124.8	2740	82.0	22,600	8.30	14.7	
E2a	201.5	4440	80.0	—	7.50	15.0	
E3sd	202.7	4460	82.0	—	8.70	15.0	22.0
E6s(new)	260.0	5720	103.0	—	9.00	15.0	
H8sb	268.3	5900	106.0	—	7.50	15.0	
H8b	268.5	5860	106.0	—	7.50	15.0	
E2d	191.3	4200	76.0	—	8.80	15.0	
3000	239.8	5290	106.0	44,100	8.40	16.0	
585	226.3	4990	100.0	*	*	16.0	
958	254.2	5600	113.0	—	8.47	16.3	
Coates.	224.6	4950	85.0	—	8.80	18.2	
K4s	359.3	7900	114.0	—	7.50	19.0	
929	326.4	7200	123.5	*	*	19.9	
535	228.0	5020	104.0	38,500	7.50	18.8	
K2as	310.7	6820	129.0	—	8.30	20.9	
K2as(stkir)	300.4	6600	122.0	—	8.10	20.9	
K2	312.0	6960	125.0	—	9.00	20.9	
K29s	386.0	8500	120.0	—	7.60	21.9	

* CANNOT BE EXTRAPOLATED DUE TO LIMITED RANGE OF TESTS.

Table 51

Data for Evaporation Curves

Engine Designation	Equival't Grate Sq. Ft.	Coal per Hour, lb		Equivalent Hourly Evaporation, lb.		Tube Length Ft.	Firing Rate lb coal per sq. ft. E.G. per hr.
		Total	Per Sq. Ft. Grate	Total	Per lb. of Coal		
Schen.*1	92.1	1290	75.0	—	8.75	11.5	
" 2	92.1	1290	75.0	—	9.30	11.5	
" 3	85.9	1200	70.0	—	9.70	11.5	
628	134.9	1890	65.0	14,900	7.90	12.0	
H6a	207.1	2880	59.0	24,100	8.40	13.7	
H6b	206.4	2890	59.0	—	8.90	13.8	
E6s (old)	241.2	3380	61.0	—	10.40	13.7	
E6	256.2	3560	65.0	—	10.80	14.0	
734	190.8	2660	79.0	23,100	8.70	14.9	
2512	124.8	1740	52.0	16,200	9.30	15.0	
E2a	201.5	2810	51.0	—	9.30	15.0	
E3sd	202.7	2840	52.0	—	10.20	15.0	
E6s (new)	260.0	3650	66.0	—	10.80	15.0	14.0
H8sb	268.3	3760	68.0	—	9.30	15.0	
H8b	268.5	3720	68.0	—	9.30	15.0	
E2d	191.3	2880	48.0	—	10.20	15.0	
3000	239.8	3350	68.0	33,000	9.90	16.0	
585	226.3	3160	64.0	29,300	9.30	15.9	
958	254.2	3550	72.0	—	9.68	16.3	
Coates.	224.6	3140	53.0	—	10.90	18.2	
K4s	359.3	5050	73.0	—	10.70	19.0	
929	326.4	4560	78.0	38,600	8.50	19.9	
K2as	310.7	4350	82.0	—	10.40	20.9	
K2	312.0	4400	79.0	—	10.90	20.9	
K2as (stk'r)	300.4	4210	77.0	—	9.90	20.9	
K29s	386.0	5400	76.0	—	9.50	21.9	
Schen.*1	92.1	1650	96.0	—	8.30	11.5	
" 2	92.1	1650	97.0	—	8.80	11.5	
" 3	85.9	1560	85.0	—	9.20	11.5	
628	134.9	2420	85.0	18,500	7.65	12.0	
H6a	207.1	3730	76.0	28,300	7.60	13.7	
H6b	206.4	3720	76.0	—	8.40	13.7	
E6s (old)	241.2	4330	78.0	—	9.55	13.7	
E6	256.2	4600	83.0	—	9.80	14.0	
734	190.8	3430	101.0	25,600	7.45	14.9	
2512	124.8	2240	67.0	18,600	8.20	14.7	
E2a	201.5	3630	65.0	—	8.40	15.0	
E3sd	202.7	3650	67.0	—	9.40	15.0	
E6s (new)	260.0	4670	84.0	—	9.90	15.0	18.0
H8sb	268.0	4840	87.0	—	8.25	15.0	
H8b	268.5	4850	88.0	—	8.40	15.0	
E2d	191.3	3440	62.0	—	9.50	15.0	
3000	239.8	4630	65.0	39,600	9.20	16.0	
585	226.3	4075	83.0	38,300	9.40	16.0	
958	254.2	4590	92.0	—	9.00	16.3	
Coates.	224.6	4050	68.5	—	9.80	18.2	
K4s	359.3	6500	94.0	—	9.95	19.0	
929	326.4	5860	101.0	*	*	19.9	
535	228.0	4100	85.0	31,900	7.80	18.8	
K2as	310.7	5600	105.0	—	9.45	20.9	
K2as (stk'r)	300.4	5400	100.0	—	9.95	20.9	
K2	312.0	5620	101.0	—	10.00	20.9	
K29s	386.0	6950	99.0	—	8.45	21.9	
Schen.*1	92.1	2030	117.0	—	7.70	11.5	
" *2	92.1	2030	117.0	—	8.40	11.5	
" *3	85.9	1890	111.0	—	8.80	11.5	
628	134.9	2970	103.0	21,700	7.30	12.0	
H6a	207.1	4550	93.0	29,300	6.50	13.7	
H6b	206.4	4560	93.0	—	7.80	13.8	
E6s (old)	241.2	5300	96.0	—	8.60	13.7	
E6	256.2	5610	102.0	—	8.80	14.0	
734	190.8	4200	124.0	30,000	7.10	14.9	
2512	124.8	2740	82.0	22,600	8.30	14.7	
E2a	201.5	4440	80.0	—	7.50	15.0	
E3sd	202.7	4460	82.0	—	8.70	15.0	22.0
E6s (new)	260.0	5720	103.0	—	9.00	15.0	
H8sb	268.3	5900	106.0	—	7.50	15.0	
H8b	268.5	5860	106.0	—	7.50	15.0	
E2d	191.3	4200	76.0	—	8.80	15.0	
3000	239.8	5290	106.0	44,100	8.40	16.0	
585	226.3	4990	100.0	*	*	16.0	
958	254.2	5600	113.0	—	8.47	16.3	
Coates.	224.6	4950	85.0	—	8.80	18.2	
K4s	359.3	7900	114.0	—	7.50	19.0	
929	326.4	7200	123.5	*	*	19.9	
535	228.0	5020	104.0	38,500	7.50	18.8	
K2as	310.7	6820	129.0	—	8.30	20.9	
K2as (stk'r)	300.4	6600	122.0	—	8.10	20.9	
K2	312.0	6960	125.0	—	9.00	20.9	
K29s	386.0	8500	120.0	—	7.60	21.9	

* Cannot be extrapolated due to limited range of tests.

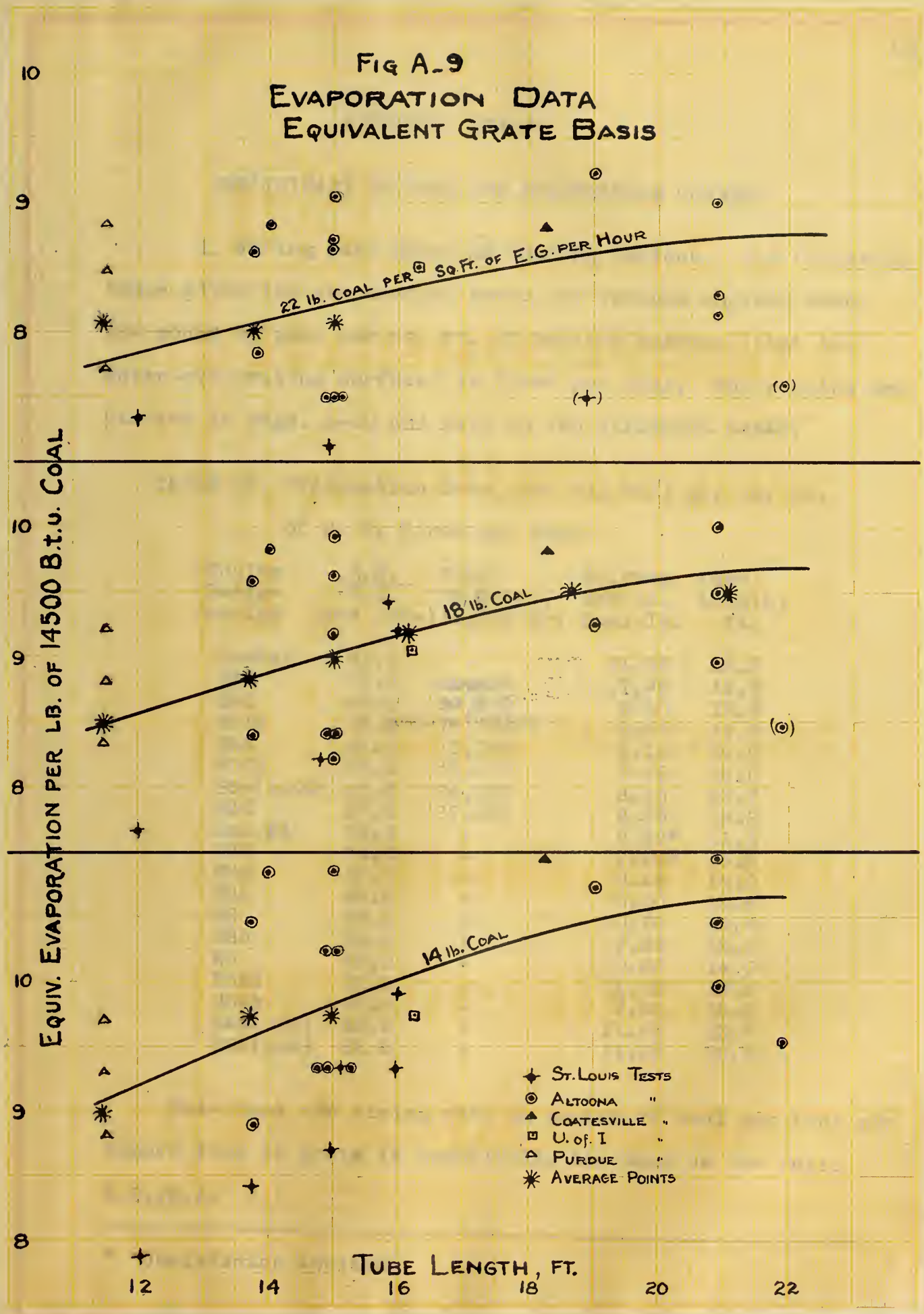


Fig A-9
Evaporation Data
Equivalent Grate Basis
22 lb. Coal per Sq. Ft. of E.G. per Hour
18 lb. Coal
14 lb. Coal
Equiv. Evaporation per Lb. of 14500 B.t.u. Coal
Tube Length, Ft.
10
9
8
12
14
16
18
20
22
St. Louis Tests
Altoona "
Coatesville "
U. of I "
Purdue "
Average Points

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APPENDIX VII-F

PRELIMINARY STUDIES FOR EVAPORATION CURVES.

1. Firing Rate Based on Heating Surface. The following table gives the evaporation rates for various engines when one pound of coal per sq. ft. of heating surface (that is, water-evaporating surface) is fired per hour. The results are plotted in Figs. A-10 and A-11 on two different bases.

TABLE 52, Evaporation Data, One lb. Coal per sq. ft. of H. S. Fired per Hour.

Engine Designation	H.S./G.A. (see note)	Total Eq.Evap., lb.per hr.	Eq.Evap. per lb. Coal,lb.	Tube Length, ft.
Coates.	51.5	-	11.05	18.2
734	85.0	22,800	7.95	14.9
585	63.0	29,600	9.10	15.9
2512	49.0	15,700	9.55	14.7
628	47.0	13,200	8.10	12.0
3000	68.0	33,000	9.70	16.0
H6a(1499)	57.0	24,300	8.55	13.7
535	67.0	27,000	8.35	18.8
Sch.#1	78.0	-	8.30*	11.5
958	74.0	-	9.60*	16.3
E2a	47.0	-	9.55	15.0
H6b	58.0	-	9.00	13.8
K2	83.0	-	10.70	20.9
H8b	69.0	-	9.30	15.0
E6	65.0	-	10.80	14.0
E3sd	37.0	-	11.10	15.0
H8sb	55.0	-	9.90	15.0
E6s(old)	48.0	-	11.05	13.7
E6s(new)	52.0	-	11.50	15.0

Note that the firing rate in pounds of coal per hour per square foot of grate is numerically the same as the ratio H.S./G.A.

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* Coal-factor applied.

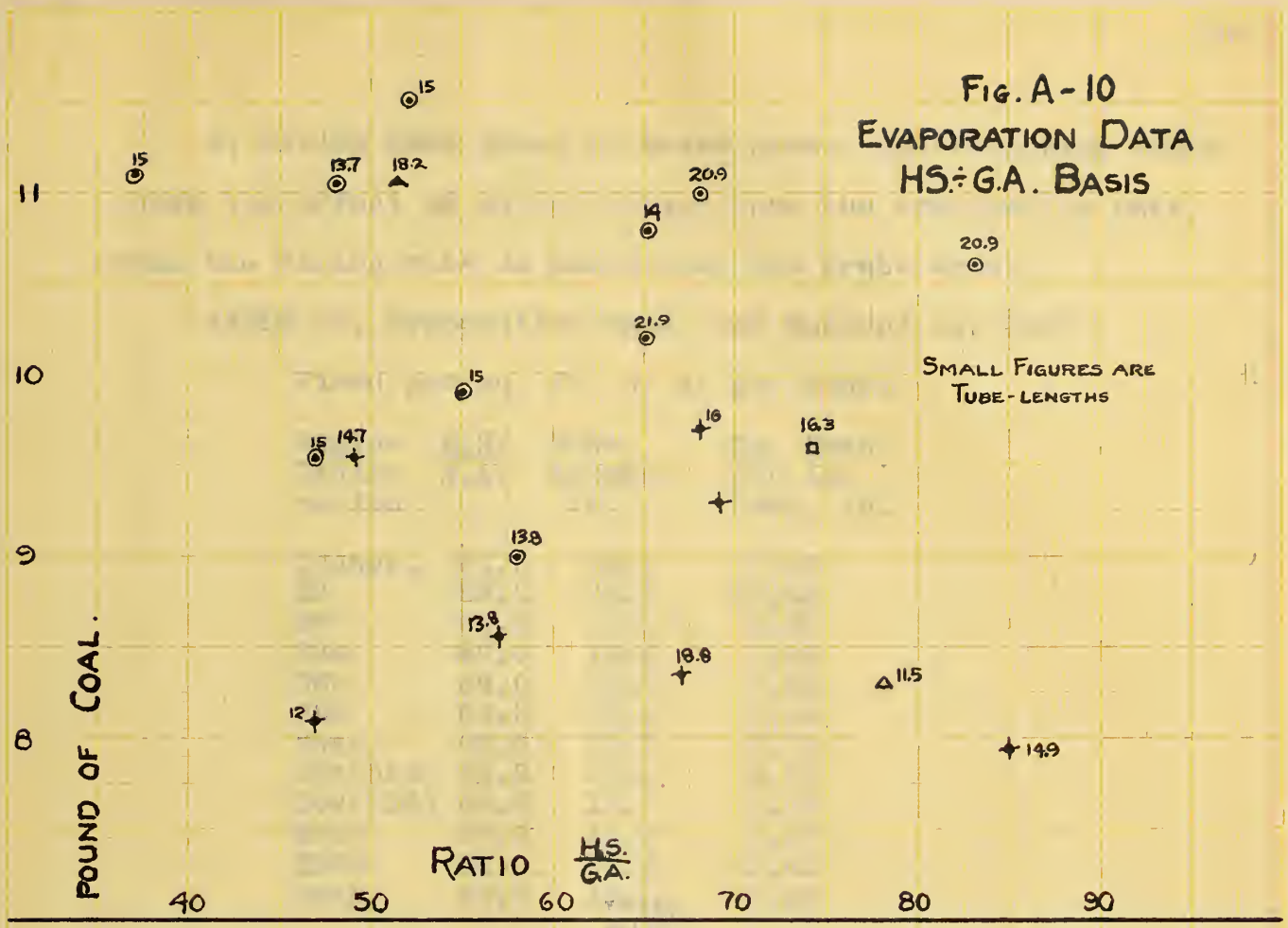
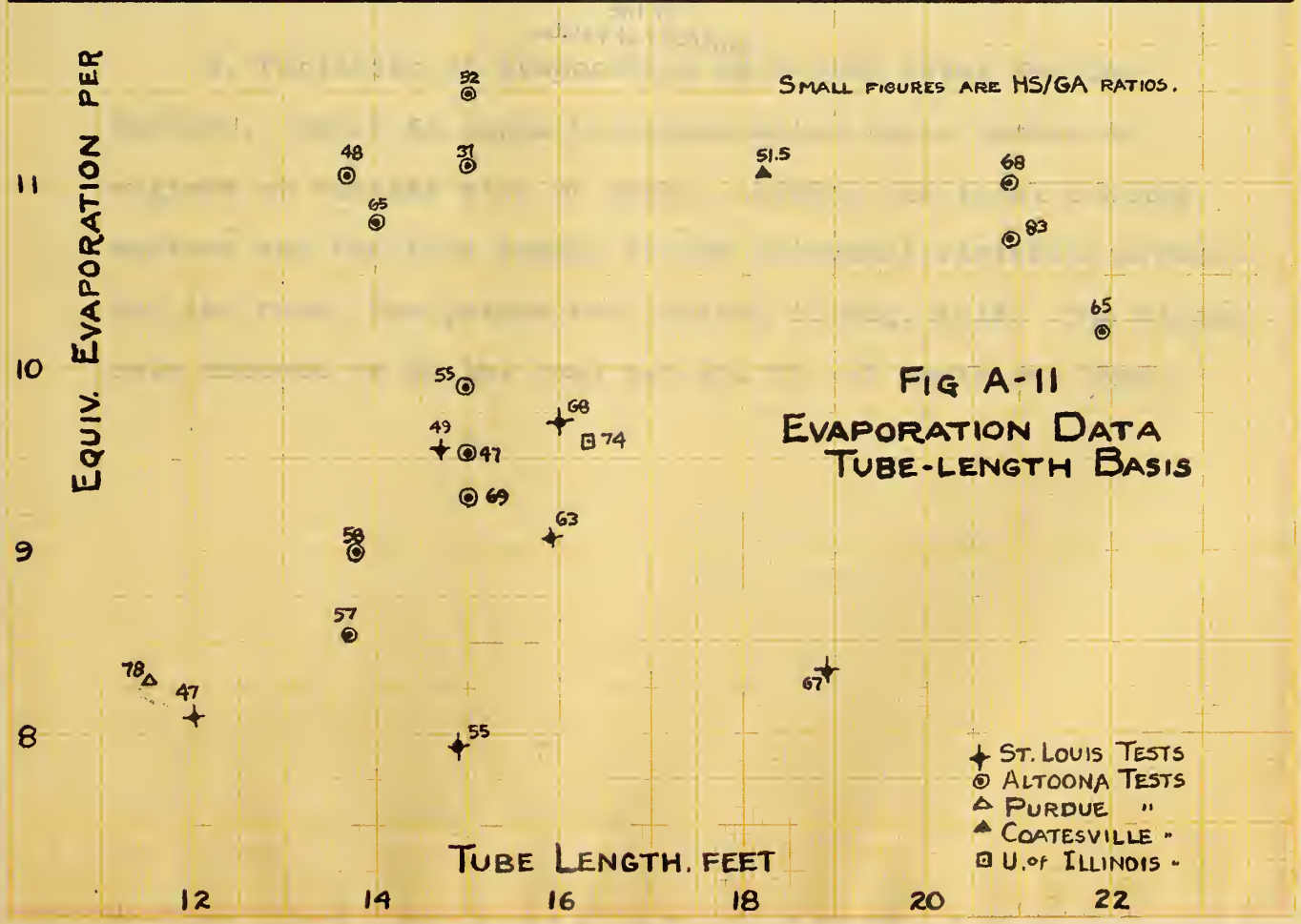


Fig. A-10
Evaporation Data
HS.÷G.A. Basis
Small Figures are Tube-lengths
Pound of Coal.
Ratio HS./G.A.
11
10
9
8
40
50
60
70
80
90
15
15
13.7
18.2
20.9
14
20.9
21.9
15
16
16.3
15
14.7
13.8
13.8
18.8
11.5
12
14.9


Small figures are HS/GA ratios.
Fig A-11
Evaporation Data
Tube-length Basis
Equiv. Evaporation per
Tube Length, feet
11
10
9
8
12
14
16
18
20
22
52
48
31
51.5
68
65
83
65
55
68
49
47
74
69
63
58
57
78
47
67
55
St. Louis Tests
Altoona Tests
Purdue "
Coatesville "
U. of Illinois "

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2. Firing Rate Based on Grate Area. The following table shows the effect of boiler design upon the evaporation rate, when the firing rate is based upon the grate area.

TABLE 53, Evaporation Data, One Hundred lb. Coal Fired per sq. ft. G. A. per Hour.

Engine Designation	H.S. / G.A.	Tube Length, ft.	Eq. Evap. per lb. Coal, lb.
Coates.	51.5	18.2	7.80
K2	83.0	20.9	10.05
E6	65.0	14.0	8.90
E2a	47.0	15.0	6.25
H6b	58.0	13.7	7.64
H8b	69.0	15.0	7.80
K2as	68.0	20.9	9.70
E6s(new)	52.0	15.0	9.10
E6s(old)	48.0	13.7	8.45
E3sd	37.0	15.0	7.50
K29s	65.0	21.9	8.45
H8sb	55.0	15.0	7.50

3. Variation of Evaporation Rate with Total Heating Surface. Table 54 shows the evaporation for a number of engines of similar size of grate, leaving the total heating surface and the tube length as the principal variables affecting the rate. The points are plotted in Fig. A-13. The firing rate assumed is 80 lb. coal per sq. ft. of grate per hour.

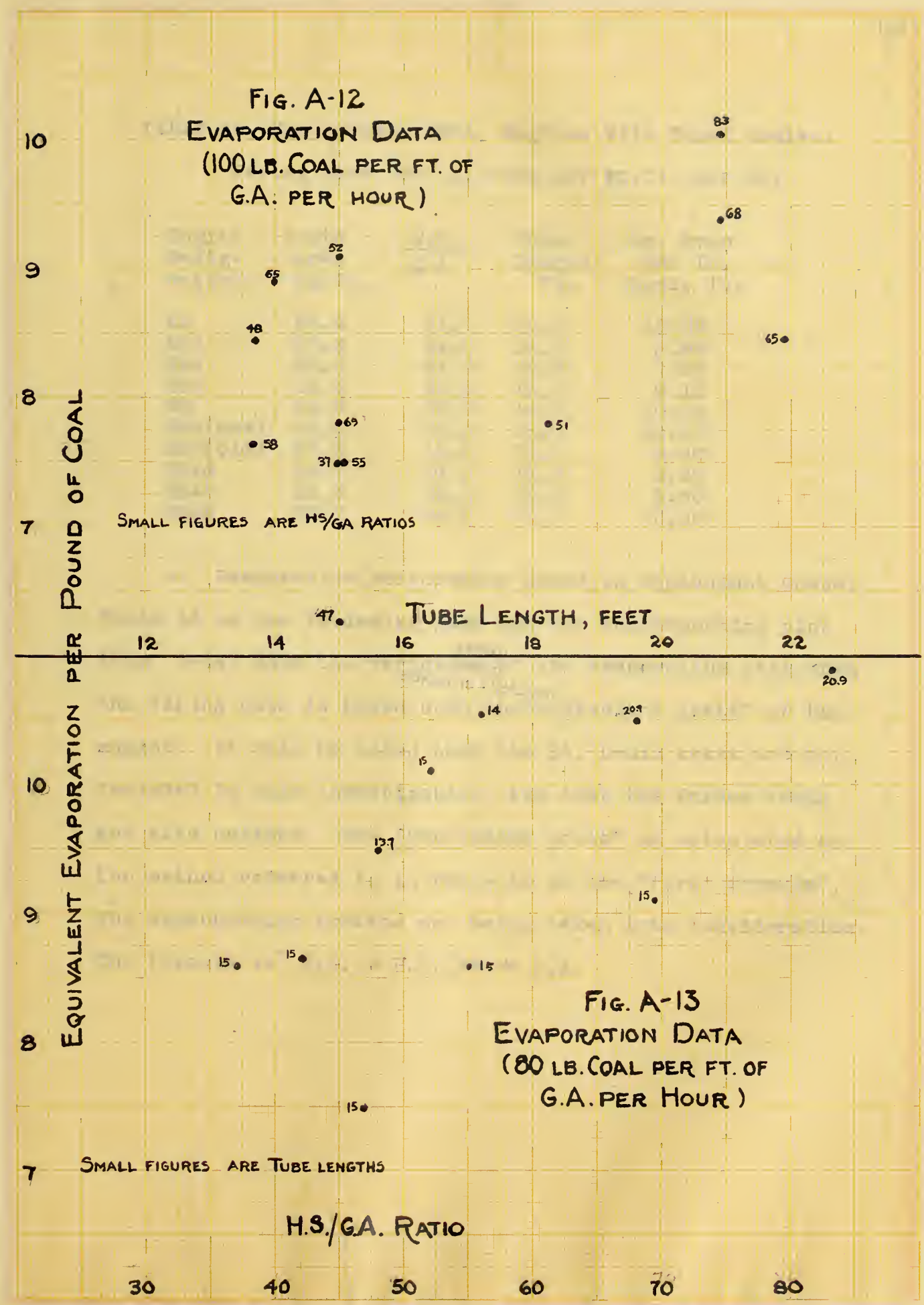
Fig. A-12
Evaporation Data
(100 lb. Coal per ft. of G.A. per hour)
Small figures are HS/GA ratios
Equivalent Evaporation per Pound of Coal
Tube Length, feet
10
9
8
7
12
14
16
18
20
22
83
68
52
65
48
65
69
51
58
37
55
47
20.9
14
20.9
15
13.7
15
15
15
15
15
Fig. A-13
Evaporation Data
(80 lb. Coal per ft. of G.A. per Hour)
Small figures are Tube lengths
H.S./G.A. Ratio
30
40
50
60
70
80

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TABLE 54, Evaporation Data, Engines with Equal Grates.

Firing Rate, 80 lb. Coal per sq.ft. per hr.

Engine Designation	Grate Area sq.ft.	H.S. / G.A.	Tube Length ft.	Eq. Evap. per lb. Coal, lb.
K2	55.4	83.0	20.9	10.90
E2d	55.3	44.0	15.0	8.65
E2a	55.5	47.0	15.0	7.50
H8b	55.3	69.0	15.0	9.10
E6	54.8	56.0	14.0	10.05
E6s(new)	55.8	52.0	15.0	10.10
E6s(old)	55.2	48.0	13.7	9.50
E3sd	54.7	37.0	15.0	8.60
H8sb	55.3	55.0	15.0	8.60
K2as	53.2	68.0	20.9	10.50

4. Evaporative Performance Based on Equivalent Grate. Table 55 on the following page and the corresponding plot (Fig. A-14) show the variation of the evaporation rate when the firing rate is based upon the"equivalent grate" of the engine. It will be noted that the St. Louis tests are not included in this investigation, and that the Purdue tests are also omitted. The "equivalent grate" is calculated by the method referred to in Table 50 as the "first formula", the superheating surface not being taken into consideration. The formula is E.G. = H.S./18 + G.A.

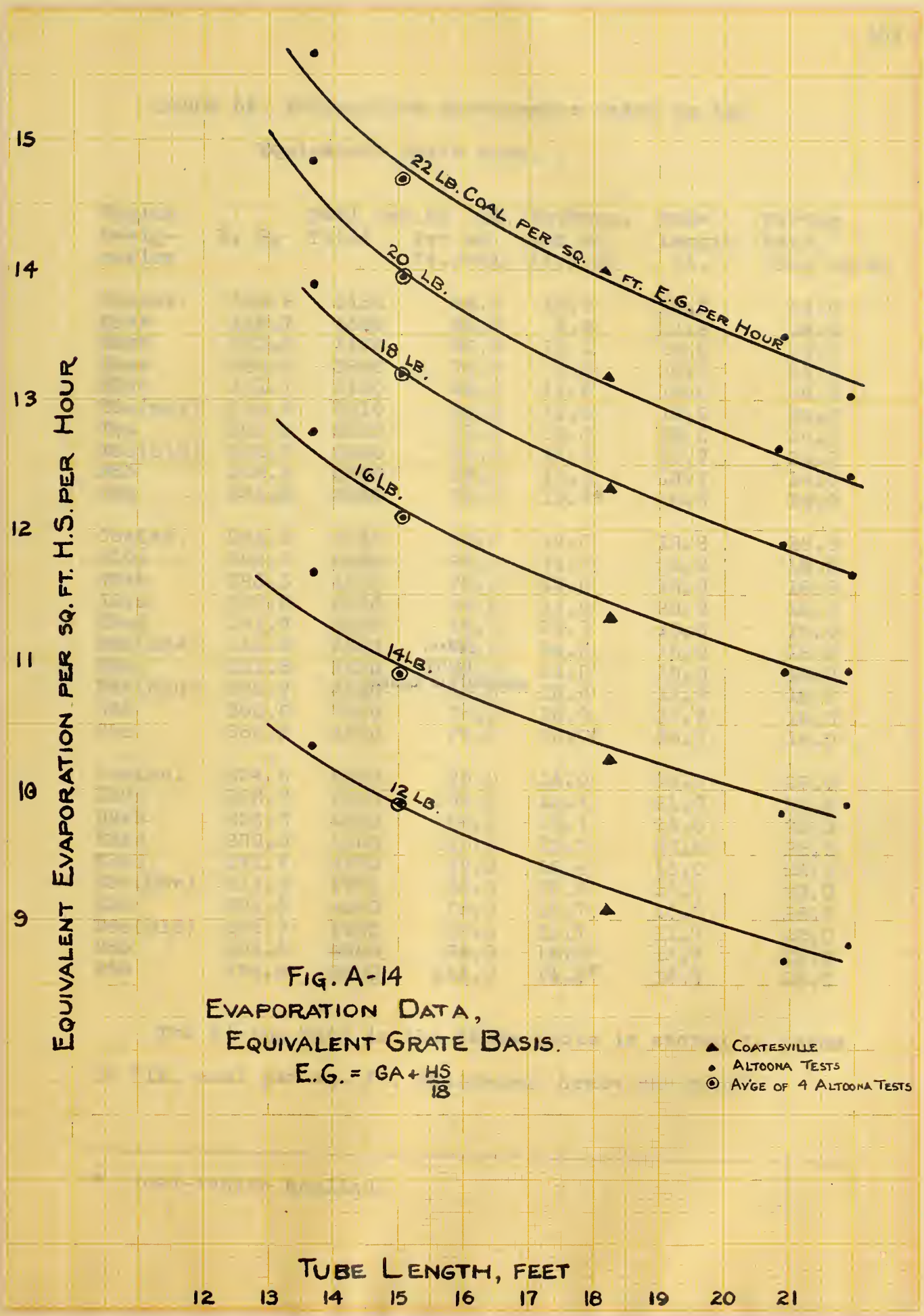

FIG. A-14
EVAPORATION DATA,
EQUIVALENT GRATE BASIS.
$E.G. = GA + \frac{HS}{18}$

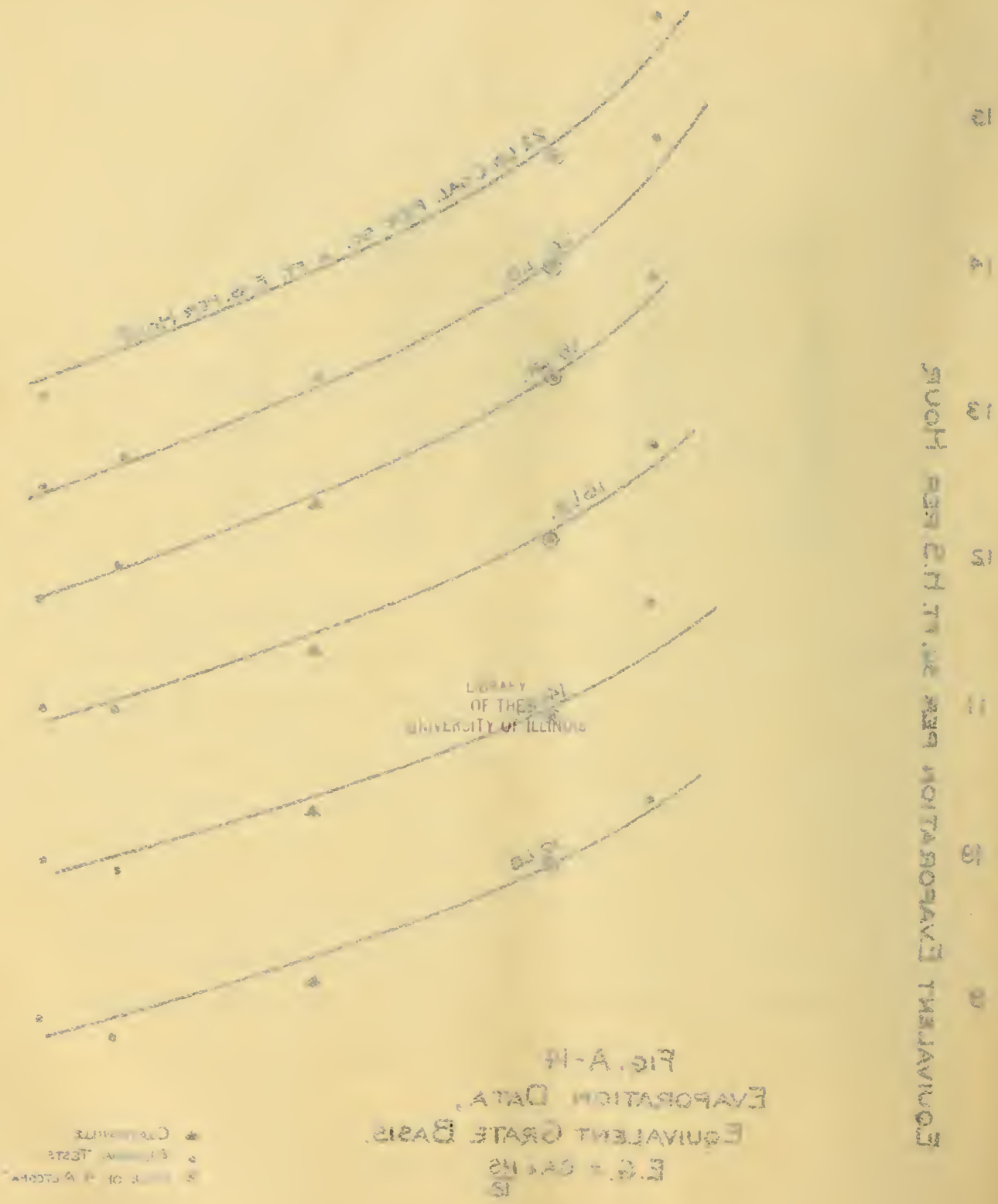

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TABLE 55, Evaporative Performance Based on the Equivalent Grate Area.

Engine Designation	E. G.	Coal per hr., lb. Total	Per sq. ft. Grate	Eq. Evap. per sq. ft. H.S.	Tube Length ft.	Firing Rate. (see note)
Coates.	224.6	3150	54.0	10.2	18.2	14.0
K29s	326.7	4560	65.0	9.9	21.9	14.0
H8sb	223.3	3120	56.0	10.1	15.0	14.0
K2sa	280.0	3920	74.0	9.8	20.9	14.0
E3sd	171.7	2400	44.0	11.6	15.0	14.0
E6s(new)	214.9	3010	54.0	12.0	15.0	14.0
E2a	201.5	2820	51.0	10.0	15.0	14.0
E6s(old)	202.7	2840	51.0	11.7	13.7	14.0
H6b	206.5	2890	59.0	10.1	13.7	14.0
958	254.2	3560	72.0	10.3*	16.3	14.0
Coates.	224.6	4050	70.0	12.3	18.2	18.0
K29s	326.7	5880	84.0	11.7	21.9	18.0
H8sb	223.3	4020	72.0	12.0	15.0	18.0
K2sa	280.0	5040	94.0	11.9	20.9	18.0
E3sd	171.7	3090	56.0	15.1	15.0	18.0
E6s(new)	214.9	3780	70.0	14.3	15.0	18.0
E2a	201.5	3630	65.0	11.5	15.0	18.0
E6s(old)	202.7	3650	66.0	13.9	13.7	18.0
H6b	206.5	3620	74.0	12.0	13.7	18.0
958	254.2	4580	93.0	12.7*	16.3	18.0
Coates.	224.6	4950	85.0	14.0	18.2	22.0
K29s	326.7	7200	102.0	13.1	21.9	22.0
H8sb	223.3	4900	89.0	13.1	15.0	22.0
K2sa	280.0	6160	115.0	13.5	20.9	22.0
E3sd	171.7	3770	69.0	16.4	15.0	22.0
E6s(new)	214.9	4730	85.0	15.9	15.0	22.0
E2a	201.5	4440	79.0	12.7	15.0	22.0
E6s(old)	202.7	4450	80.0	15.7	13.7	22.0
H6b	206.5	4550	94.0	14.0	13.7	22.0
958	254.2	5600	113.0	14.5*	16.3	22.0

The firing rate in the above table is stated in terms of "lb. coal per sq. ft. Equivalent Grate per hour".

---

* Coal-factor applied.

5. Evaporative Performance Based on Equivalent Grate, Using Data for All Engines. In the following table the calculations of Table 55 are extended to all engines for which data are available. The evaporation is calculated in pounds per pound of coal fired instead of the former basis. No figures are given for engines using saturated steam, as they are identical with those of Table 51. The firing rate is stated as before, in pounds of coal fired per square foot of equivalent grate. Fig. A-15 shows the results of this study.

TABLE 56, Evaporative Performance, Based on Equivalent Grate.

Engine Designation	E. G. sq.ft.	Coal per hr., lb. Total	Per sq.ft. Grate, lb.	Eq. Evap. per lb. Coal, lb.	Tube Length ft.	Firing Rate lb.
628	118.6	1650	66.0	9.6	12.0	14.0
E6s(o)	202.7	2960	53.0	10.8	13.7	14.0
E6s(n)	214.9	3000	54.0	11.4	15.0	14.0
E3sd	171.7	2400	44.0	10.7	15.0	14.0
H8sb	223.3	3060	55.0	9.9	15.0	14.0
K2sa	280.0	3920	74.0	11.3	20.9	14.0
K29s	326.7	4550	64.0	10.1	21.9	14.0
628	118.6	2160	74.0	7.8	12.0	18.0
E6s(o)	202.7	3650	66.0	10.2	13.7	18.0
E6s(n)	214.9	3870	69.0	10.6	15.0	18.0
E3sd	171.7	3080	56.0	10.0	15.0	18.0
H8sb	223.3	4020	73.0	11.0	15.0	18.0
K2sa	280.0	5040	94.0	10.6	20.9	18.0
K29s	326.7	5890	84.0	9.2	21.9	18.0
628	118.6	2600	90.0	7.5	12.0	22.0
E6s(n)	202.7	4660	83.0	9.3	13.7	22.0
E6s(n)	214.9	4730	85.0	9.8	15.0	22.0
E3sd	171.7	3780	69.0	9.3	15.0	22.0
K2sa	280.0	6160	115.0	9.9	20.9	22.0
K29s	326.7	7170	101.0	8.9	21.9	22.0

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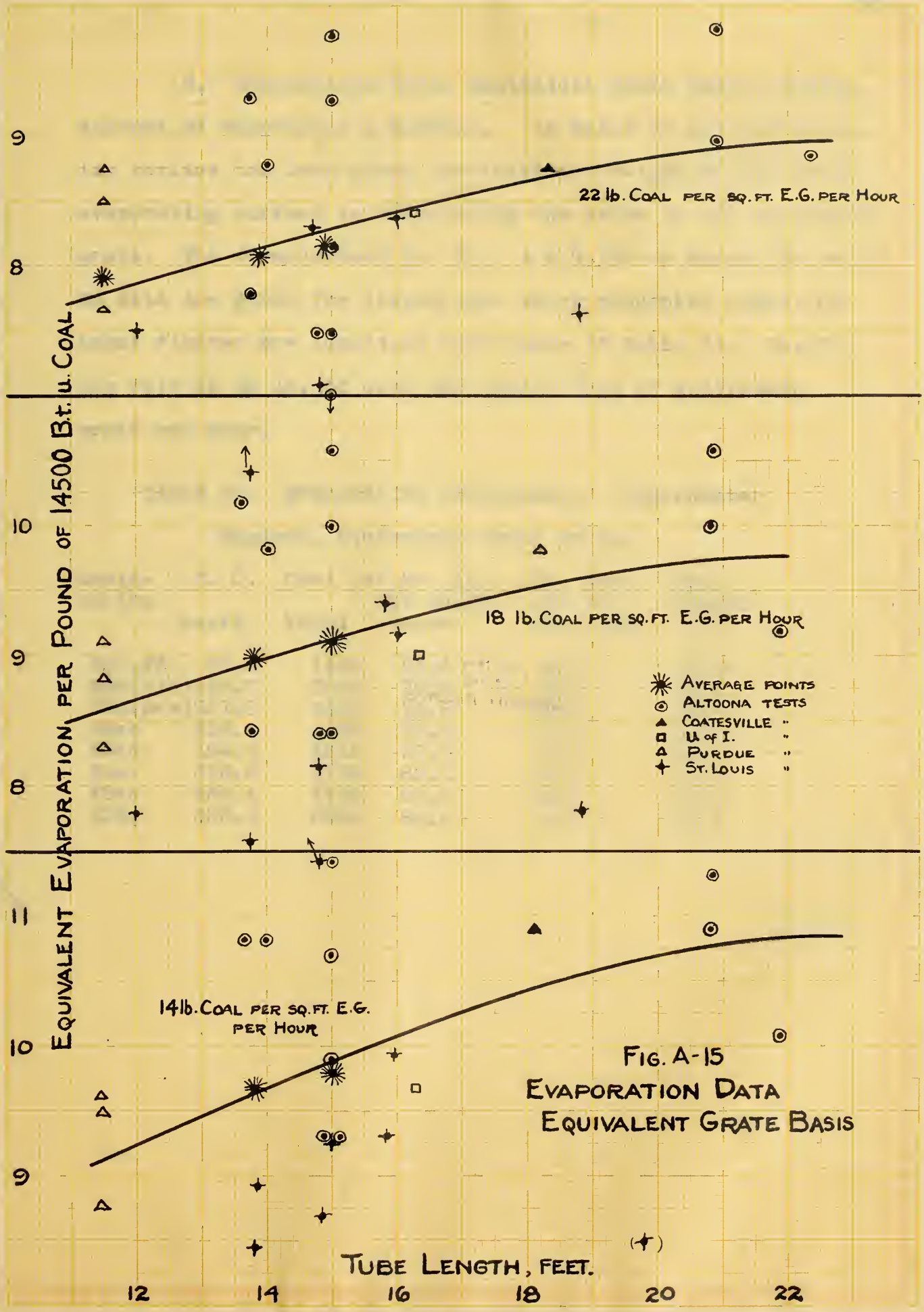

Fig. A-15
Evaporation Data
Equivalent Grate Basis

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6. Evaporation Data, Equivalent Grate Basis, Taking account of superheating surface. In Table 57 the superheating surface has been given one-third the weight of the water-evaporating surface in calculating the value of the equivalent grate. The formula used is E.G. = H.S./18 + Sup.S./50 + G.A. No data are given for locomotives using saturated steam, as these figures are identical with those of Table 51. The firing rate is 18 lb. of coal per square foot of equivalent grate per hour.

TABLE 57, Evaporative Performance, Superheater Engines, Equivalent Grate Basis.

Designation	E. G. sq.ft.	Coal per hr., lb. Total	Coal per hr., lb. Per sq.ft. Grate	Eq. Evap. per lb. Coal, lb.	Tube Length ft.
Sch.#3	78.8	1420	83.5	9.2	11.5
E6s(old)	216.7	3900	70.5	9.9	13.7
E6s(new)	231.3	4180	75.5	10.3	15.0
H8sb	239.8	4330	78.8	8.8	15.0
E3sd	184.0	3310	60.9	9.7	15.0
K4s	316.9	5700	82.1	10.4	19.0
K2sa	265.4	4790	89.5	10.1	20.9
K29s	335.4	6020	85.3	9.1	21.9

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